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AUTHORITY

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WADC TECHNICAL REPORT 53-180
PART II
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STUDY OF MINIATURE ENGINE-GENERATOR SETS

Part II. Investigation of Engines, Fuels, and Lubricants

RICHARD G. SALTER
MARION L. SMITH
CHARLES P. ROBERTS

THE OHIO STATE UNIVERSITY RESEARCH FOUNDATION

DECEMBER 1954

Statement A
Approved for Public Release

WRIGHT AIR DEVELOPMENT CENTER

NOTICES

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EQUIPMENT LABORATORY
CONTRACT No. AF 18(600)-192
PROJECT No. 6058

WRIGHT AIR DEVELOPMENT CENTER
AIR RESEARCH AND DEVELOPMENT COMMAND
UNITED STATES AIR FORCE
WRIGHT-PATTERSON AIR FORCE BASE, OHIO

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FOREWORD

This report was prepared in the Mechanical Engineering Department of The Ohio State University under Contract No. AF 18(600)-192 "Study of Miniature Engine-Generator Sets" with The Ohio State University Research Foundation. Work was conducted under Project No. 6058, "Electrical Generation Equipment," Task No. 60266, "Development of Miniature Engine-Generator Sets," (formerly RDO No. 656-2112), and was directed by the Equipment Laboratory, with Dr. Edwin Naumann serving as project engineer.

This report covers the basic engine investigation phase of the project, conducted during the period from June 1953 through December 1954, and is a continuation of the work reported in WADC TR 53-180. Generator investigations performed during this period are still in progress and are not included in this report. Problems pertinent to complete engine-generator sets will be studied during the remaining one and one-half years of the proposed investigation program.

This report is the second part of WADC Technical Report 53-180. The first part of this report, dated May 1953, was published under the basic report number only; it should be considered as Part I although it was not so marked.

ABSTRACT

Information concerning the design and performance of various types of miniature internal combustion engines is presented. Comparisons are given for the performance of miniature and larger engines with means for extrapolating large engine practice into the miniature size-range. Performance curves and design features of selected miniature test engines are presented and discussed. Desirable design features for various types of applications are given. A summary is presented of the fuel and lubricant characteristics necessary for miniature engines. This report describes a continuation of the earlier study presented in WADC TR 53-180.

PUBLICATION REVIEW

The publication of this report does not constitute approval by the Air Force of the findings or the conclusions contained therein. It is published only for the exchange and stimulation of ideas.

FOR THE COMMANDER:

for S. T. SMITH
Colonel, USAF
Chief, Equipment Laboratory
Directorate of Laboratories

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INTRODUCTION

Miniaturization and weight reduction of mechanical and electronic equipment have become increasingly important over the past decade. This is particularly true of equipment with intended military or airborne applications where size and weight are of major logistic and economic importance. Simultaneously there has been a tremendous increase in the development of electrical and electronic devices for communication, detection, guidance, and a host of other applications, with promise of unparalleled growth in this field in the future.

For the remote or the portable electronic device, it is necessary to provide a self-sufficient electrical power supply capable of producing the desired type of electrical power, in the necessary amounts, and for the required length of time. Extreme reliability of the power supply is often essential, since failure could result in jeopardizing lives and equipment.

The power supplies, of course, are subject to the size and weight limitations imposed on all portable or remote equipment, and a continuing effort is being made to develop lighter, smaller, and more efficient supply systems.

Self-sufficient devices requiring small amounts of electrical power of from 35 to 400 watts have so far been dependent upon batteries, particularly where a reliable source is necessary. Batteries, however, are relatively heavy, bulky devices supplying only direct current, and the necessity for their use frequently compromises the desired performance of the end product. As an alternative, the miniature engine-generator set appears to offer tremendous possibilities if the performance and reliability of such machines can approach the results obtained with larger engine-generator sets.

At the outset of the project reported here, very little technical information was available concerning the miniature engine-generator set or the design and performance of its components. In particular, the manner in which miniature component design critieria differ from those developed for components of larger size was unknown.

This report, which is the second of the over-all study, summarizes part of an investigation, underway during the past $2\frac{1}{2}$ years, of miniature engine-generator sets. It deals with the study of the design and performance of miniature engines, keeping in mind that the effort thus far invested in the miniature engine phase of the investigation has been sufficient only to provide a greater understanding of the problems involved and to establish general orders of magnitude of the design and performance criteria of miniature internal-combustion engines. The performance data set forth do not necessarily represent the ultimate of which miniature engines are capable, since time was not available for complete development of any particular engine configuration or even for the consideration of many configurations.

In addition to the design, construction, and testing of several selected miniature engines, an effort was made to glean as much information as possible from the literature, from manufacturers of commercial engines and components, and from modification and testing of various commercial miniature engines. As time would not allow intensive investigation into all phases of miniature engine design and performance, an effort was made to concentrate on those areas where initial studies would be most advantageous.

WADC TR 53-180 describes the effort during the first year of the proposed four-year program, and includes the long-range development program which has been pursued during the period covered by this report, June 1953 through December 1954. Phase I of the development program, dealing with engine investigation, has been completed and the results are described here. Phase II, the generator investigation, is in progress and results will be presented in the next technical report.

Some of the various problems of Phase III which pertain to combined engine-generator sets will be studied during the next one and one-half years of the proposed program. These include development of satisfactory systems for starting, cooling, carburetion, silencing, speed and voltage regulation, etc. Additional studies of altitude and cold temperature performances, design criteria for intake and exhaust systems, packaging, and similar problems related to combined units will be conducted.

SECTION I

SUMMARY

The results of this investigation have served to define, in general terms, the practicability of the various types of engines and the orders of magnitude of the design and performance criteria associated with miniature engines at the present state of the art.

As in any engineering problem, selection of the engine type and of design characteristics likely to provide an engine having the most satisfactory performance for all of the anticipated operating conditions necessitates a carefully considered compromise in which all factors of the intended type of application are included. Classifying the possible applications into two groups representing short and long periods of continuous operation, it is possible to make general recommendations based on project experience. For applications requiring only short periods of operation, weight of equipment is more significant than weight of fuel consumed, and it appears that the superior prime mover would be a high-speed, high specific-power, two-cycle engine. For this application, glow ignition would provide a good, relatively simple ignition system, whereas spark ignition would give slightly superior performance and some additional complications. The free-breathing, cross-scavenged design would be best for the short operating periods. A two-cycle design using some form of loop scavenging, and affording slightly better fuel economy, would be the best design where several hours of continuous operation would be required.

For longer periods of operation, in excess of 25 to 50 hours, the weight of fuel and lubricant used becomes relatively more important with respect to total equipment weight, and a moderate-speed, moderate-specific power, four-cycle engine, because of its better economy, appears to be the best choice of prime mover. The specific weight of the four-cycle engine, in pounds per brake horsepower, will be approximately 1.2 to 2.0 times that of a two-cycle engine. The improved fuel economy gained with the four-cycle design will provide a fuel saving over a long operating period which will more than compensate for the increased equipment weight. Spark ignition is the only satisfactory ignition system for use with the four-cycle engines.

Unusual configurations which add further complication to the engine design are considered to be unsatisfactory for most engine-generator applications. These designs include such configurations as the opposed-piston and poppet-valve, uniflow-scavenged two-cycle engines, and exhaust supercharger valves for assymmetrical timing of two-cycle engines. To date, compression ignition also has proven unsatisfactory in the miniature size range.

Octane rating of the fuel has very little effect on the performance of miniature engines using either a spark or a glow plug ignition system. Fuel volatility likewise appears to be a relatively unimportant factor if the fuel evaporates. Engine starting characteristics

are closely correlated with fuel volatility. With their higher heating values, petroleum fuels are superior to the alcohol fuels for engine-generator applications, since the resulting fuel consumption on both a weight and a volume basis is considerably lower and engine operation and reliability are not affected. Standard petroleum fuels may be used satisfactorily in most applications of engine-generator sets.

Experience has shown that operating life of the engine may exceed several hundred hours. Substantial improvements in reliability have been realized, but additional development work is required with the problems attendant to combinations of engines and generators, such as reliability of starting, adequate cooling, satisfactory control systems, part-load performance, and performance under various altitude and temperature conditions.

SECTION II

THE MINIATURE ENGINE

1. GENERAL

1.1 INTRODUCTION

There is no precedent to define the general size range of internal-combustion engines that can be termed miniature. There will be a considerable latitude of opinion as to the sizes that should be included.

For the purpose of this report, the range to be discussed is to be inferred from the original program which called for a study of miniature engine-generator sets having output capacities of 35 to 400-watts. The miniature engine then might be defined arbitrarily as the size necessary to drive, at sea level, a 35 to 400 watt generator plus all necessary accessories (cooling fan, ignition system, governor, etc.), using only 60 per cent of its maximum rated sea level power.

Considering reasonable values of generator efficiency and accessory power requirements, the rated maximum outputs would vary roughly from 0.2 to 2.25 bhp. For this type of application, it is reasonable to assume that specific power outputs up to 2.0 bhp per cubic inch displacement could be achieved. Assuming an average value of 1 bhp per cubic inch results in a cylinder displacement of 0.2 to 2.0 cubic inches. The engine weight range would be from 0.2 to 2.0 lb.

Prior to this investigation, miniature engines of the size range defined above were available commercially only in the model industry. There were a multitude of model engines of many types available with rated speeds ranging from 7000 to 18,000 rpm. These engines were developed for a very specialized application which did not stress the need for reliability, endurance, and economy as much as for high specific power output and very low manufacturing cost. Engines of 3.0 to 9.0 cubic inches displacement and rated speeds ranging up to 6000 rpm, which can be classed as small engines, were found as prime movers for numerous devices such as generators, lawn mowers, bicycles, chain saws, etc.

The majority of the experimental work of this investigation was conducted in the area of the miniature higher-speed engines. Less experience had been accumulated in this area, and applications stressing reduction of weight and size could benefit most from investigations with these units.

As previously noted, many of the problems associated with miniature engines are not due to the size range but rather are common to all size engines. However, certain of these problems are more

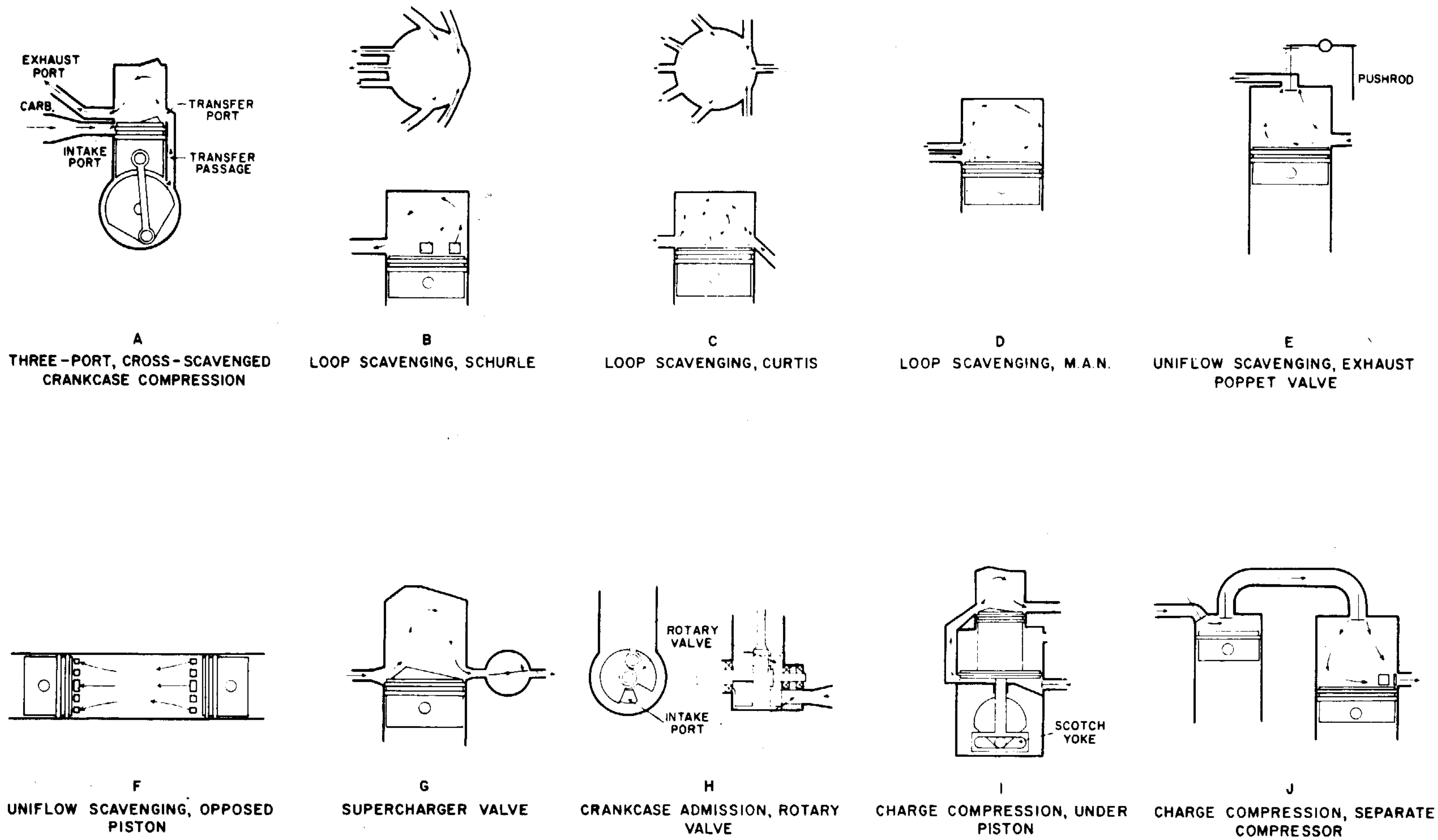


FIG. I SAMPLE TWO-CYCLE ENGINE CONFIGURATIONS - SCHEMATIC REPRESENTATION critical with smaller size, since at comparable mean piston speeds the rotating speeds are considerably higher for the miniature engine. Periodic cycle functions, such as induction, scavenging, valve actuation, etc., take place at a considerably higher frequency, and the time available for each function is very much reduced. Hence, difficulty can be expected where the increased frequency results in a reduction of the efficiency with which the function is performed. Further, the miniature engine has a higher ratio of combustion chamber surface to volume, resulting in an increase in the energy lost to the cooling system per cycle. The result is a tendency to operate at a thermal efficiency lower than that of larger similar engines.

1.2 MINIATURE ENGINE FEATURES

Conventional reciprocating internal-combustion engines can be divided into the two major internal-combustion-engine categories, two-stroke cycle and four-stroke cycle. Further subdivision can be made according to the ignition and thermodynamic processes (Otto or Diesel cycle) the mechanical configuration, the type of scavenging employed, cooling systems, etc.; the combinations possible are almost unlimited although experience has proven that certain combinations are preferable to others.

1.2.1 Two-Stroke Cycle Engine

The two-cycle engine offers great potential as a prime mover for miniature engine-generator sets. The simpler forms of the two-cycle engine require few moving parts and are capable of very good specific power outputs. There are, however, implications of inherently poor efficiency associated with the simple two-cycle scavenging process.

Figure 1 shows schematic representations of some of the more common two-cycle-engine mechanical configurations. These sketches represent only a small percentage of the configurations possible, and the discussion that follows does not cover all of the details of operation. A more complete discussion of engine configurations and details of operation can be found in many internal-combustion engine references and textbooks, for example Refs. 1-3.

The simplest two-cycle engine configurations, shown in part (A) of Figure 1, is the three-port, cross-scavenged, crankcase-compression, two-cycle engine. The mixture is inducted from the carburetor through the piston-controlled intake port into the crankcase where it is compressed by the piston as it descends on its power stroke. When the transfer port is uncovered by the piston, the compressed mixture passes up the transfer passage, through the transfer port and into the cylinder. As the mixture fills the cylinder it displaces the residual combustion products from the previous cycle and forces them out the exhaust port. During this process there is some mixing of the intake mixture and the combustion products with a resulting loss of fuel mixture when these products are exhausted. In addition, some of the intake mixture flows from the intake port directly out the exhaust port without contributing to the scavenging process; this process is known as short-circuiting. The deflector shown on the piston serves to deflect the mixture toward the cylinder head to promote more complete scavenging and to reduce short-circuiting. Compression of the cylinder charge takes place and ignition occurs at a point prior to the top-dead-center piston position. As the piston descends on the power or expansion stroke the exhaust port is uncovered before the transfer port. This allows for a "blow-down" period necessary to reduce the pressure in the cylinder sufficiently to allow the fresh intake mixture to enter the cylinder when the transfer port is uncovered.

In this engine all three valves for flow control (intake, transfer, and exhaust) are piston-controlled ports symmetrically timed in the cycle. That is to say, opening and closing of these ports will occur at identical piston positions on downward and upward strokes and consequently at the same angles before and after top- or bottom-center crank position. This condition does not lend itself to versatility of flow control and is thus a most inefficient valving system. For example, since the exhaust port must open before the transfer port, to allow for blow-down, it will close after the transfer port on the upward movement of the piston; this allows loss of induction mixture through the exhaust port, and reduction of cylinder pressure and the power that would be obtained with higher initial cylinder pressure.

Parts (B), (C) and (D) of Figure 1 show three systems of cylinder-port configurations designed to improve the efficiency of cylinder scavenging by inducing "loop" flow. By these methods the mixture is directed up and generally away from the exhaust ports, thus forcing it to travel in a loop, and in so doing, to scavenge the farthest regions of the combustion chamber. In general, these methods employ flat-topped pistons and simple combustion chamber shapes, but more complicated transfer passage and transfer port configurations. With a given port timing it is generally not possible to obtain as large port areas as can be obtained with the simple cross-scavenging system shown. The M.A.N. system shown in part (D) is the least satisfactory of the three types of loop scavenging for miniature engines.

Parts (E) and (F) of Figure 1 show two methods of uniflow scavenging which result in single-direction flow of both the intake mixture and the combustion products, an even more efficient scavenging process. In addition both of these methods allow for asymmetrical timing of scavenging events, i.e., exhaust before transfer. However, both systems add the complication of additional mechanisms, a poppet valve and its necessary actuation linkage in configuration (E) and an additional piston, connecting rod, crankshaft, and interconnecting linkage in configuration (F).

Part (G) of Figure 1 shows another means of providing asymmetrical port-timing. A rotary valve is inserted in the exhaust passage to close it off before the transfer port closes. The rotary valve, however, absorbs considerable power thus tending to offset the improved scavenging efficiency.

Methods for providing for asymmetrical timing of the crankcase admission generally involve some type of rotary valve, a popular type of which is shown in part (H) of Figure 1. This type of valve involves the use of some additional mechanism requiring very accurate construction to minimize the leakage past the valve when it is closed without unduly increasing the drag and therefore the driving power required. Another type of rotary valve system employs a hollow crankshaft with a radial port located between the two main bearings on the crankshaft. With this type it is somewhat difficult to provide adequate flow areas while maintaining a reasonable crankshaft diameter. A pressure-operated reed valve is another type used for crankcase admission, particularly where flexibility of engine speed is desired. This valve works like a check valve, allowing flow into the crankcase when the pressure in the crankcase is less than the inlet pressure and closing to seal the crankcase when the crankcase pressure exceeds the inlet pressure. There is no fixed timing associated with this valve, and thus more flexible engine speed is possible. A slight reduction in engine specific power can be expected to accompany the pressure loss associated with the operation of the reed valve.

A basic disadvantage of the crankcase compression engine, as will be discussed later, is the necessity for adding lubricating oil to the fuel. A means of obviating this necessity as well as increasing the scavenging pressure is shown in part (I) of Figure 1. The method shown utilizes a scotch yoke in place of the crank to obtain a simple reciprocating motion of the piston rod, thereby making possible a seal between the piston rod and the crankcase. A crosshead mechanism can be also used to achieve this result. Compression takes place in the cylinder immediately under the piston and the crankcase can then be lubricated by conventional methods. Cylinder lubrication can be accomplished by metering oil to the piston or cylinder wall.

Separate compression can also be used to supply the pressurized scavenging mixture as illustrated in part (J) of Figure 1. This, of course, allows complete flexibility in selection of the quantity and pressure of the scavenging mixture.

1.2.2 Four-Stroke Cycle Engine

The classification of four-cycle engine configurations is generally accomplished by specifying the types of valves and valve actuation systems employed. This discussion will be limited to the poppet valve which so far has proven to be the most practical type. Valve placement and orientation in the combustion chamber are usually of the overhead-valve (valve-in-head) type or the "L"-head (side-valve) type. Actuation of the valve is accomplished by means of a rotating cam. The overhead-valve system has become more popular in recent years since it allows the designer to use higher compression ratios and more efficient breathing passages than are possible with the L-head configuration. Actuation of the overhead-valve system requires a push rod and rocker arm system if the cam is located near the center line of the crankshaft. To reduce reciprocating weight for high-speed operation, an overhead cam is employed.

For a complete discussion of the four-cycle engine construction and operating principles, attention is again directed to the many texts and reference books in the field, a few of which are listed in the references of this report.

1.2.3 Comparison of Miniature Two and Four-Cycle Engines

The following comparison of miniature two- and four-cycle engines is derived not only from the inherent design characteristics involved, but also from accumlated experience with both miniature and larger engines. Many of the features mentioned here will be considered in more detail later.

1.2.3.1 Power Output Experience has shown that the simple, miniature, port-scavenged two-cycle engine is inherently capable of a higher specific power output than is the four-cycle engine. The most important factors involved are, of course, the increased number of power cycles per revolution in the two-cycle, together with its inherent capability of operating at higher rotating speeds. The latter factor stems from the simplicity of the piston ports as compared to the poppet-valve gear train of the four-cycle. The accelerations involved establish a maximum practical limit to the frequency of poppet-valve gear action with present techniques. For miniature engines this limit appears to be of the order of 6000 valve functions per minute (12,000 rpm). This means that the maximum piston speeds will become seriously restricted as the displacement is reduced. Below 10,000 rpm the two- and four-cycle engines do not differ widely in specific power, but the two-cycle maintains a slight superiority because its BMEP is usually slightly greater than one half that of the four-cycle.

The two-cycle must precompress the mixture to provide the necessary scavenging potential, whereas the four-cycle utilizes positive displacement scavenging. In general, the latter requires less power. A factor tending to counteract this effect is the friction power required to drive the four-cycle valve gear.

1.2.3.2 Fuel Economy The quantity of fuel burned per cycle in a miniature engine is so small that direct fuel injection into the cylinder is impractical. Thus, it is necessary to carburete the fuel into the induction mixture, which, in the two-cyqle, serves also as the scavenging medium. In the two-cycle scavenging process it is impossible to prevent mingling of the scavenging mixture and the combustion products, and inevitably some fuel will be lost by short circuiting, even in the better designed scavenging systems. In large two-cycle diesels where the fuel is injected directly into the cylinder, the loss of short-circuited scavenging air represents only a loss of the pumping power expended on that air.

In contrast, the scavenging process of a four-cycle engine, particularly when designed for constant speed, can be very closely controlled for excellent efficiency with only a small loss of fuel mixture. Consideration of these facts leads to the conclusion that a well-designed miniature four-cycle engine inherently will be capable of better fuel economy than the miniature two-cycle, a conclusion which experience substantiates.

In the crankcase-compression two-cycle engine, the lubricating oil, however introduced, has free passage to the combustion chamber through the transfer system. In addition to other detrimental effects, this results in an oil consumption several times larger than occurs in the four-cycle engines.

1.2.3.3 Reliability From a mechanical standpoint the simplest form of reciprocating internal combustion engine is the three-port two-cycle engine as shown in part (A) of Figure 1. The only moving parts of such an engine are the piston, connecting rod, and crankshaft. It is obvious that the probability of mechanical engine failure will be increased when these parts are coupled with additional mechanisms as in the more complicated two-cycle engines and the four-cycle engine. There are, however, other factors of importance when considering overall reliability.

The small four-cycle engine has long held the reputation of being easier to start and of giving more stable operation than the two-cycle. The tests performed in the present investigation have supported this view. The miniature four-cycle engines start easier than the two-cycles and are apparently less critical of mixture adjustment for smooth, stable operation. A well-tuned two-cycle engine seems to be somewhat more prone to unexplainable speed fluctuations. The presence of such instability would seem to suggest possible transient combustion problems stemming from the less positive scavenging process and a possibility of dynamic instability in the inherently more complicated induction system. Although time has not permitted a detailed investigation of these stability problems, it is felt that successful development of a four-cycle engine having stability of operation and ease of starting will present fewer difficulties than will be encountered with two-cycle designs.

Another problem found in two-cycle engines with piston-controlled exhaust ports is that carbon deposits form in the exhaust ports. This condition is considerably aggravated by the high oil consumption of the crankcase-compression engines. The oil deposited on the exhaust ports each cycle is subjected to the hot exhaust products flowing out the ports. The resulting carbon deposits grow from the top edge of the port, which is first uncovered by the piston, downward and around the edge of the port until the port is almost completely closed off. This not only reduces the exhaust port area but also changes the exhaust timing; the combination seriously reduces the power output and efficiency. For the average small industrial engine, recommended maintenance procedures call for periodic cleaning of the ports, the average nominal operating time being of the order of 25 to 75 hours. A discussion of some of the problems of lubrication and combustion of small two-cycle engines can be found in Ref. 22.

Ignition systems are another source of potential unreliability. The two types of systems in general use are conventional spark ignition and glow plug ignition. Compression ignition has not proven very satisfactory for miniature engines. The glow plug utilizes a tungsten filament wire which is electrically heated for starting and warm-up, after which it is kept hot by the combustion. In general, the glow plug is successful only when frequency of combustion is kept above a certain value of the order of 6000 cycles per minute. Lower frequencies of combustion allow time for too much cooling of the filament between cycles, and the filament temperatures fall below the minimum required for reliable ignition. This limitation restricts the glow plug to two-cycle engines exclusively. Spark ignition, on the other hand, can be used with either two- or four-cycle engines and can be operated successfully by either magneto or battery systems at ignition frequencies up to 18,000 cycles per minute or at as low a speed as desired.

The spark and glow plugs present certain problems when used for long periods. Commercial glow plugs, in general, will remain electrically continuous for only a relatively short period of operation, often for only 8 hours of operating time. There is, however, apparently no limitation as to how long they will function if the engine operation is continuous. If the engine is stopped after an extended running period, it is usually necessary to install a new glow plug that can be electrically heated for starting.

The miniature commercial spark plugs of the 1/4 inch and 3/8 inch thread sizes have shown tendencies to develop leaks between the center electrode and the insulation. With some additional development, however, very reliable miniature spark plugs should be feasible. Depending on the fuel-lubricant combination, compression ratio, combustion chamber shape, etc., there is a varying tendency for both spark and glow plugs to become fouled. Test results show almost complete inconsistency on this score and it is felt that the best combinations for a particular engine will be found only by specific development work with that engine.

1.2.4 Preliminary Design Considerations

When a specific need for a miniature engine-generator set arises and the complete spectrum of operating requirements and conditions has been established, an analysis will be necessary to provide the "preliminary design" specifications for the engine involved. The "preliminary design" should define, in general, the engine type, cycle, size, speed, ignition, fuel, etc. Following this "preliminary design", a "prototype detail design" will be made and one or more experimental models will be built. As with all sizes and types of internal combustion engines, an experimental development program will have to precede the final production design so that many engine parameters can be optimized for the specific unit involved. This program could include studies of compression ratio, ignition and valve timing, operating temperature, combustion chamber shape, carburetion, throttling and governing, and general structural modification. The extent to which these parameters are studied and optimized will depend upon the application on the engine and the economic desirability of such studies. Past experience forms the basis for the selection of the range of these parameters to be used in the prototype design.

The data and recommendations given in this report are intended primarily as an aid to more accurate selection of engine types, cycles, sizes, etc., for the "preliminary design", as well as of the various values of the parameters necessary for the prototype design. They can not be expected to obviate the necessity for the development phase which must follow.

Certain general considerations involved in making the "preliminary design" analysis are outlined below. For convenience, these considerations will be termed application factors. It should be again emphasized that the majority of these items are not peculiar to miniature engines and will be well known to those familiar with larger engine practice. The application factors of primary importance to the preliminary design are as follows:

a. Total service life
b. Type of duty (time and load utilization factors)
c. Design speed and power output
d. Degree of reliability necessary
e. Physical characteristics (weight, volume, shape, durability)
f. Environmental conditions
g. Fuels and lubricants desired
h. Speed and load control necessary (accuracy, stability, and response)
i. Operational attitude

Also, depending on the application, other factors of importance include:

j. Initial cost (number of units and manufacturing ease)

k. Storage
l. Degree of self sufficiency, or accessibility for servicing
m. Starting system (manual or automatic)
n. Operating ease and safety
o. Maintenance cost and accessibility
p. Electrical and acoustical noise levels
q. Balance and vibration
r. Exhaust removal

The preliminary design arrived at will depend not only on these application factors but also on the relative emphasis given to them; it will be selected by matching these factors with the advantages of the different engine cycles and with performance criteria such as mean piston speed, BMEP, economy, reliability, etc.

1.3 COMPARATIVE PHYSICAL CHARACTERISTICS

In comparing the physical characteristics of the miniature engine with those of the larger commercially available engines, the following relations will be considered: (a) specific power (maximum brake horsepower per pound weight) versus engine size (maximum brake horsepower per cylinder); (b) economy (minimum brake specific fuel consumption) versus engine size; and (c) utilization (maximum brake mean effective pressure) versus piston speed.

The parameter chosen here to represent engine size is but one of several in common use, all of which are in some respects inadequate. However, brake horsepower per cylinder is believed to be an appropriate description of size, considering the engine-generator application under investigation.

The other physical characteristics mentioned previously were chosen for comparison because of their value in the selection of suitable power plants for an engine-generator application.

Another important factor, which cannot be extensively compared here because of insufficient data in the literature, is service life. It is generally known that the service life of the miniature engine is shorter than that of the larger type of engine. There are several reasons for this difference. In the past, there has been a high probability of structural failure in the miniature engine after a short period of operation, because of inadequate mechanical design. The inherent high rotational speed of the miniature engine does not necessarily indicate a high probability of failure, inasmuch as the corresponding bearing and structural loads are comparable to those of larger engines operating at equivalent piston speeds and can be compensated for by proper mechanical design. Piston speeds of many miniature engines are relatively low, thereby insuring good piston, ring, and cylinder life, assuming adequate cooling and lubrication.

In the miniature two-cycle engine there is at present the problem of carbon deposits which drastically limit the duration of continuous operation, at least as compared to the standards of other engines having lubrication independent of the fuel mixture.

Statistical data for this section were obtained from Ref. 26, manufacturers' data, and laboratory test results.

1.3.1 Specific Power and Engine Size

Figure 2 represents performance data for American built internal combustion engines, all commercially available in 1954, compared for specific weight (engine weight per brake horsepower) on the basis of engine size (BHP/cyl.). The superiority of the miniature engine in this respect is illustrated. The aircraft engine obviously approaches the same superiority, as would be expected since the aircraft engine is also designed for light weight and high power output.

Inspection of Figure 2 further reveals that the classes of engines other than the miniature and aircraft fail to define a narrow range of specific weight. This reflects in many cases, the variety of applications to which the engines in a particular class may be subjected.

It may be concluded that engines having low specific weight can be built for practical purposes, but with some sacrifice in fuel economy or reliability or with a large development and manufacturing cost.

1.3.2 Economy

Figure 3 represents fuel economy (minimum brake specific fuel consumption) versus engine size for engines believed to represent the most economical in their respective classes. An extensive statistical study of this characteristic is not possible, for lack of published complete performance characteristics, particularly economy, of commercial engines. However, by reporting the economy performance of a few engines considered best in their respective classes on the basis of economy only, a definite trend can be established.

The trend, indicated in Figure 3, clearly shows that the larger engine has a definite advantage over the miniature engine (regardless of whether two- or four-cycle) with respect to fuel economy.

The large engine, particularly the large stationary engine, is designed especially for good economy. Such has certainly not been true of the small and miniature engine classes to date. Only recently has much effort been expended to improve the efficiency of the very small engine. Of course, the small engine is inherently less efficient than the large engine, but more improvement can be expected from the former than for the large engine which is already perfected to a high degree.

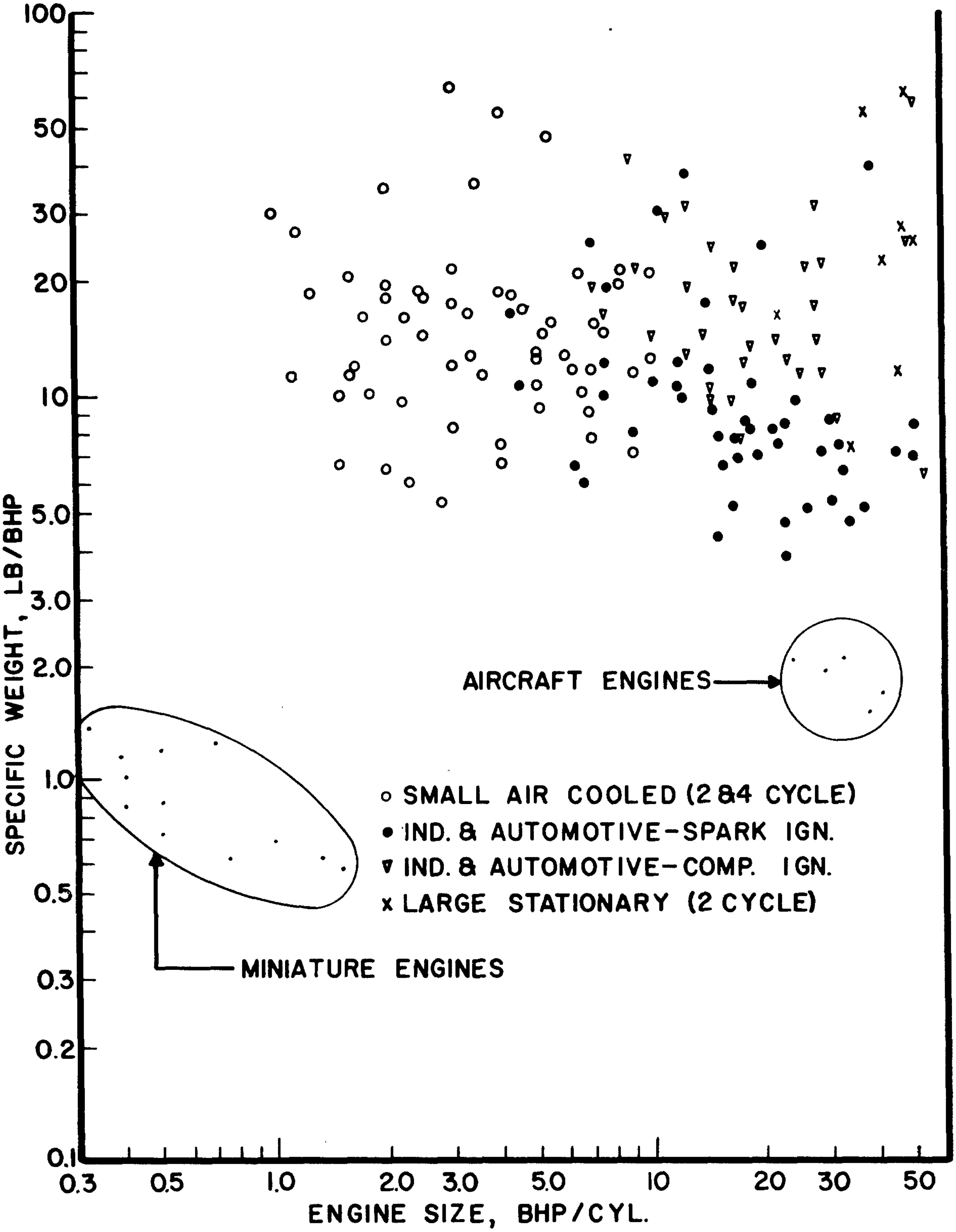


Figure No. 2. Specific Weight Vs. Engine Size for Various Commercial Engines.

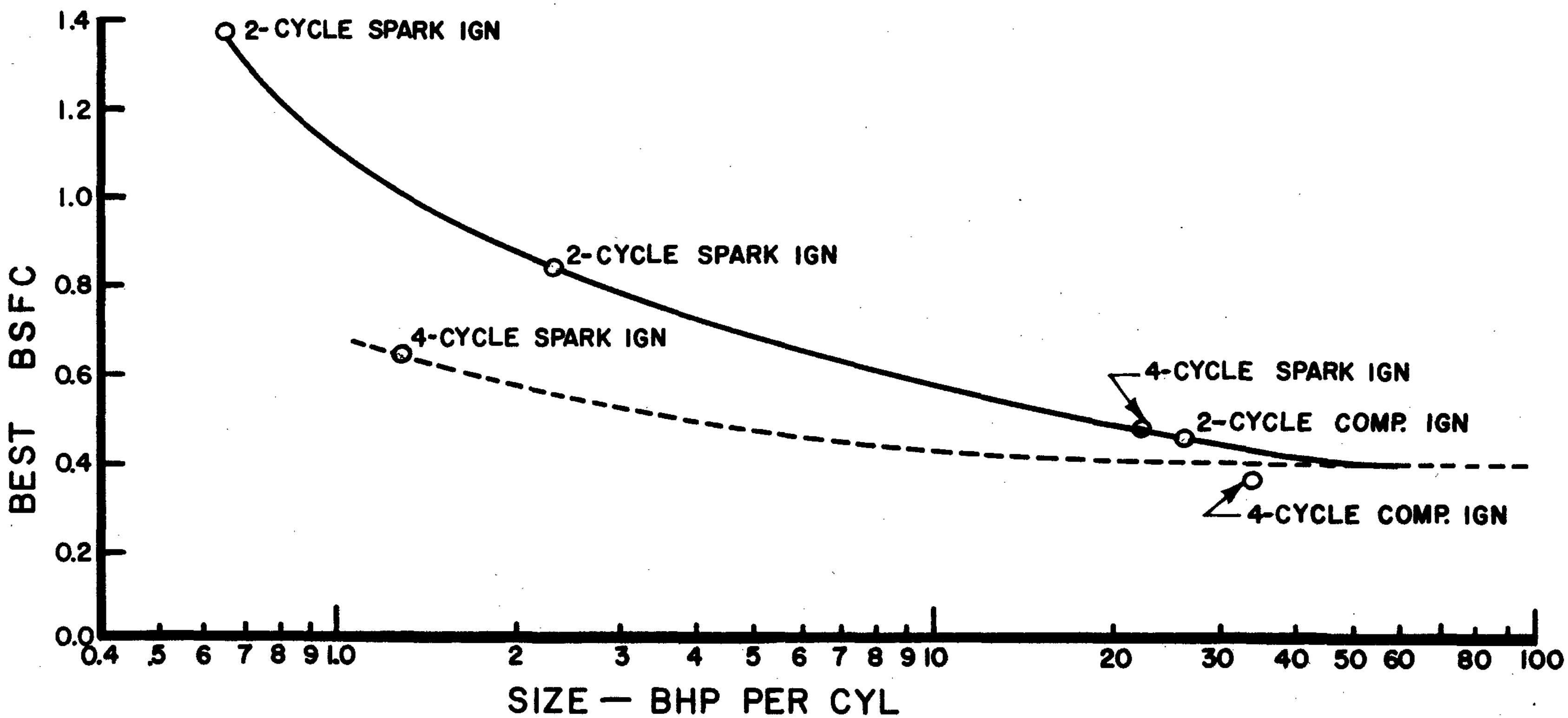


Figure No. 3. Fuel Economy Vs. Engine Size for Selected Commercial Engines

1.3.3 Utilization of Displacement

The term utilization as used here will refer to maximum brake mean effective pressure, an expression which indicates how well an engine uses its piston displacement to produce effective work. Information representative of current American engine design and relating utilization to piston speed is illustrated in Figure 4. Piston speed was chosen as a basis for comparison inasmuch as this affords a comparison independent of cylinder size, Ref. 5. Thus, engines developed for similar applications will be grouped in a narrow range on this plot. This statement is substantiated by the grouping of performance results for engines developed for transportation, such as the private automobile engine and the gasoline and diesel truck engines. The utilization of piston displacement in the automotive engines is generally greater than in the engines of the various other classes considered.

Also obvious from an inspection of Figure 4 is that the utilization in the miniature engine is very much below the average of all other engines considered, with the exception of the small air-cooled two-cycle engines. However, the piston speeds in the miniature engine category do embrace a wide range of values.

Lines representing various values of brake horsepower per square inch of piston area are also included in Figure 4 for comparison purposes.

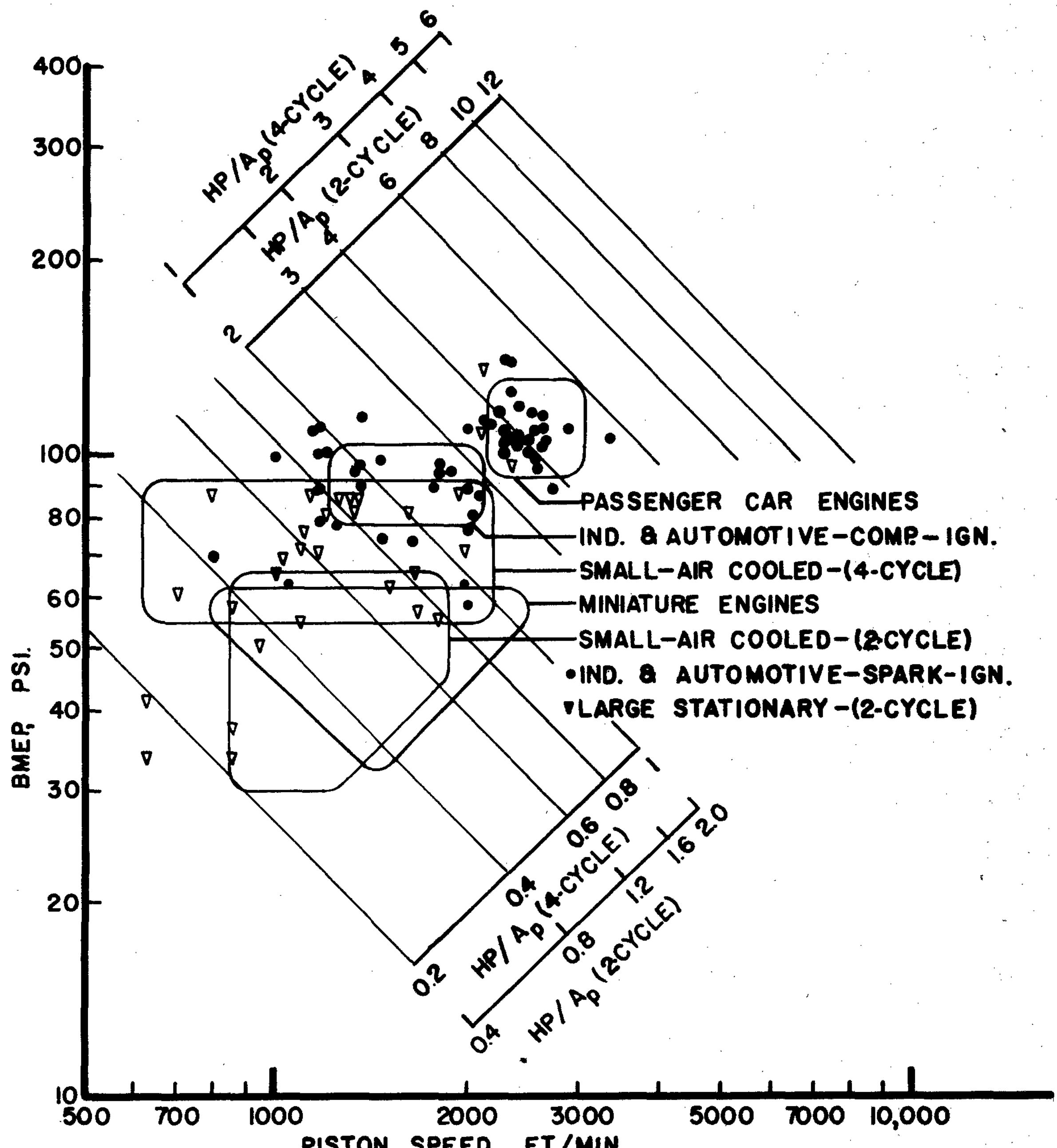


Figure No. 4. Utilization of Displacement Vs. Piston Speed for Various Commercial Engines.

2. PERFORMANCE

A discussion of the performance potential of reciprocating internal combustion engines necessarily must involve orders of magnitude rather than specific values. The reason for this is that a large number of factors affect the performance, all of which have certain practical limitations. The ultimate performance of which a given engine type is capable is related to the amount of development work that can be invested in it. Practical considerations limiting this ultimate performance include the relative importance of engine life, economy, and specific power.

Very little practical experience with miniature engine performance has been accrued to date in comparison to the background existing for larger engine sizes as exemplified by the automotive field. With this in mind, it is obvious that the performance criteria given here define the present "state of the art" for the miniature sizes and not necessarily the maximum practical potential of such engines. However, it is felt that the values given are attainable in a well-designed and developed engine. In some cases performance may exceed the representative values presented here.

2.1 POWER OUTPUT AND ECONOMY

2.1.1 Comparison Criteria

A comparison of the performance of internal combustion engines of different sizes is a very useful approach in determining the potential of miniature engines. There has been some investigation into the performance of similar engines of different size (for example Ref. 5) and there is, in general, agreement that similar engines operating at the same mean piston speeds are capable of producing comparable brake mean effective pressures. By this hypothesis it is then possible to compute the specific power potential of miniature engines by comparing them with well-developed larger engines of similar design.

There is also among engineers in the internal combustion engine field a common theory that the upper limit of specific horsepower capacity of the reciprocating engine cylinder is the limitation imposed by cylinder cooling. This would include the heat transfer characteristics of the cylinder itself and the ability of practical cooling systems to remove the transferred heat. With certain minor approximations this theory can provide another comparative hypothesis, that engines of similar design will be capable of producing the same power per unit area of cooling surface. For similar engines, however, any area or squared length parameter of the engine cylinder can be used for the comparison, i.e. bore area, $(\text{stroke})^2$, or $(\text{diam.})^2$. It is interesting to note that with some algebraic manipulation it can be shown that the two comparative hypotheses described here are identically the same.

The following series of figures are included as an aid to the use of present performance information by means of the comparative hypotheses presented above. Figures 5 through 17 are plots of specific power (BHP/cubic in.) vs. engine size (stroke, in.) at constant mean piston speed with lines of constant brake mean effective pressure (BMEP, psi). The points spotted on these curves represent the performance of various engines, as described in Table 1, at the selected piston speeds. These curves can be used to ascertain the feasibility of a given level of specific power production, by comparison with the present state of the art for the size involved and/or by extrapolation of current practice in other sizes, using the comparative hypotheses. Points A through C represent the best performance to date of the project test engines. It should be noted that these engines are not highly developed in the sense that most commercial engines are. The range of engine size was extended to include the very highly developed automotive group (points H, I, J, K, and L) for purposes of comparison. For orientation, two scales have been included at the top of Figures 5 through 17; one defining the RPM variation over the range of values of stroke, for the specific piston speed involved, and another defining the displacement variation over the stroke range at an arbitrary bore-stroke ratio (R) of 1.2. The bore-stroke ratio of 1.2 represents a mean value in the range used for miniature engines. The tabulations of points presented within Figures 5 through 17 include the bore-stroke ratio (R) of the engines as well as the BSFC, where available, which the engine achieved for the performance conditions plotted. The subscript (a) appearing after the BSFC indicates the fuel used was alcohol; all other data are for operation on petroleum fuel. Bore-stroke ratios for all other engines plotted (Nos. 1 to 52) are listed in Table 1.

Figures 18 and 19, included for convenience, show (18) the computed relationship of RPM as a function of stroke for various piston speeds and (19) stroke vs. displacement for various bore/stroke ratios.

There are several methods of approach for using these curves to aid in preliminary engine design. A comparison of the level of specific power output selected for a proposed engine design with that being used in current practice, for the size involved, can be considered as an indication of the development time necessary to obtain this performance. For example, considerable development of any new and different miniature design will be required before performance comparable to that of well-developed commercial similar larger engines can be achieved. Lower specific power can be achieved with less development; and conversely, higher specific power (if possible) will be achieved only after much additional development time.

The commercial engines for which performance data have been plotted are representative of current practice in terms of data available at the time of this report. The information can be kept up-to-date by adding

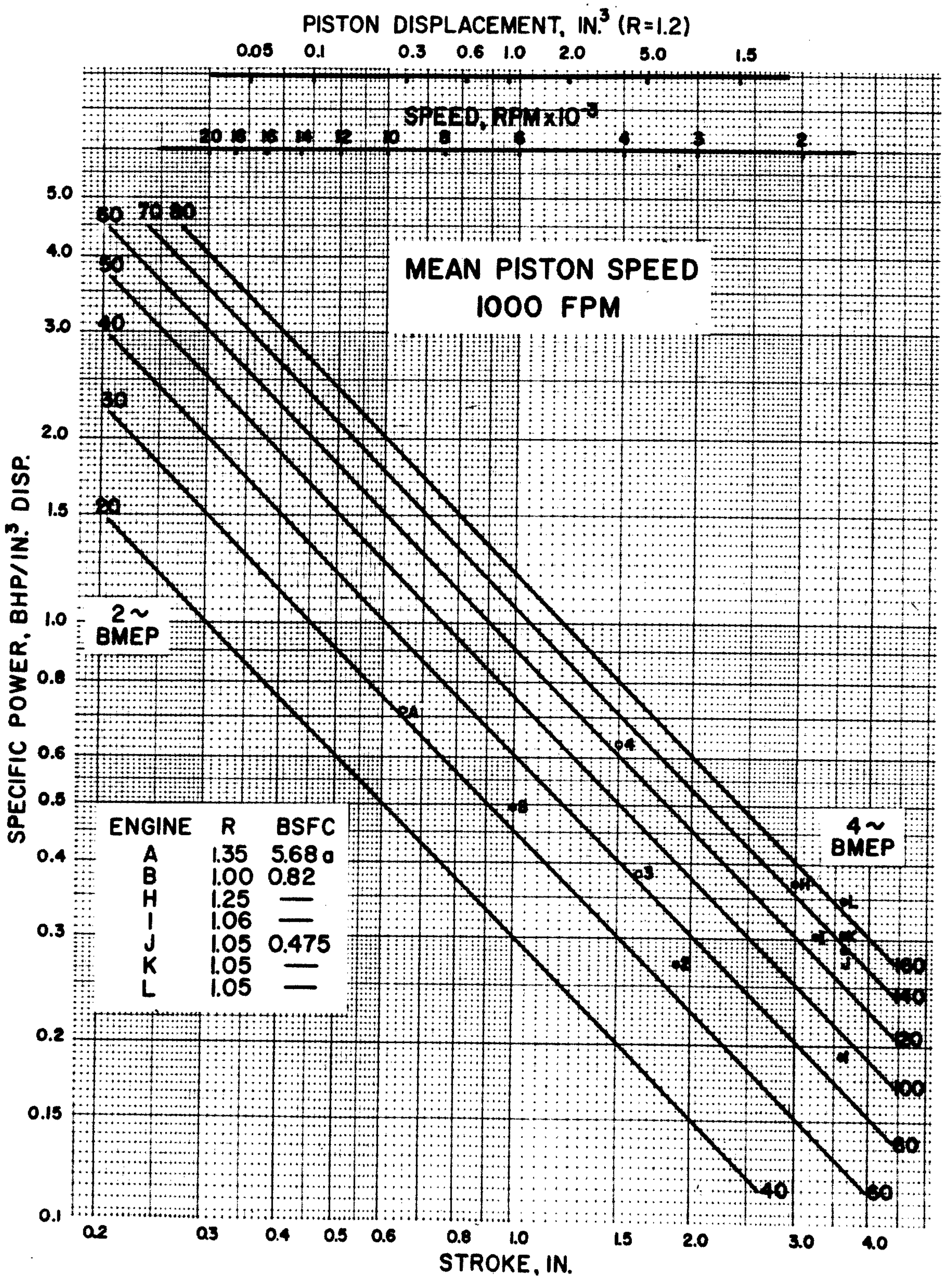


FIG. 5 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED

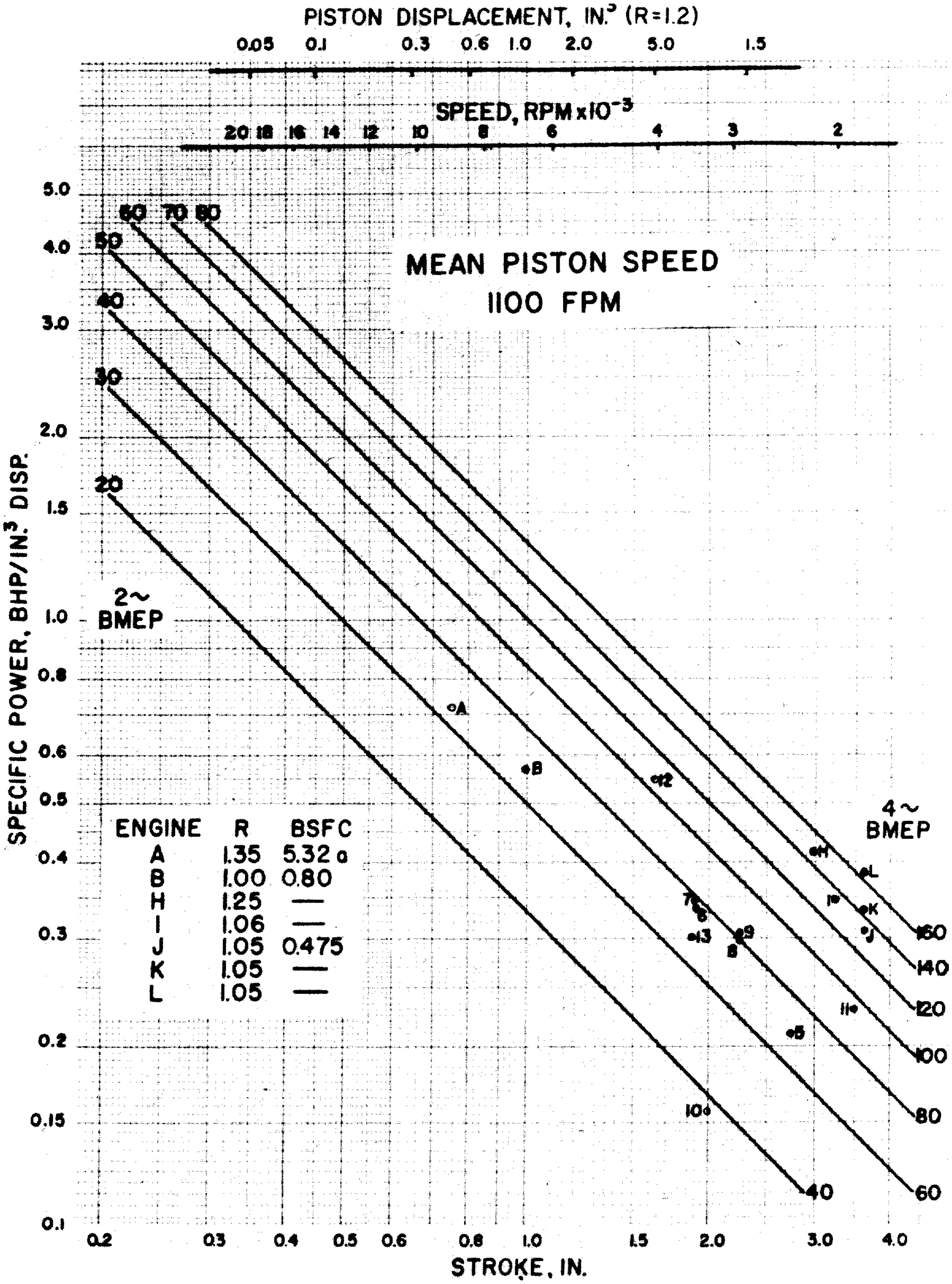


FIG.6 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED

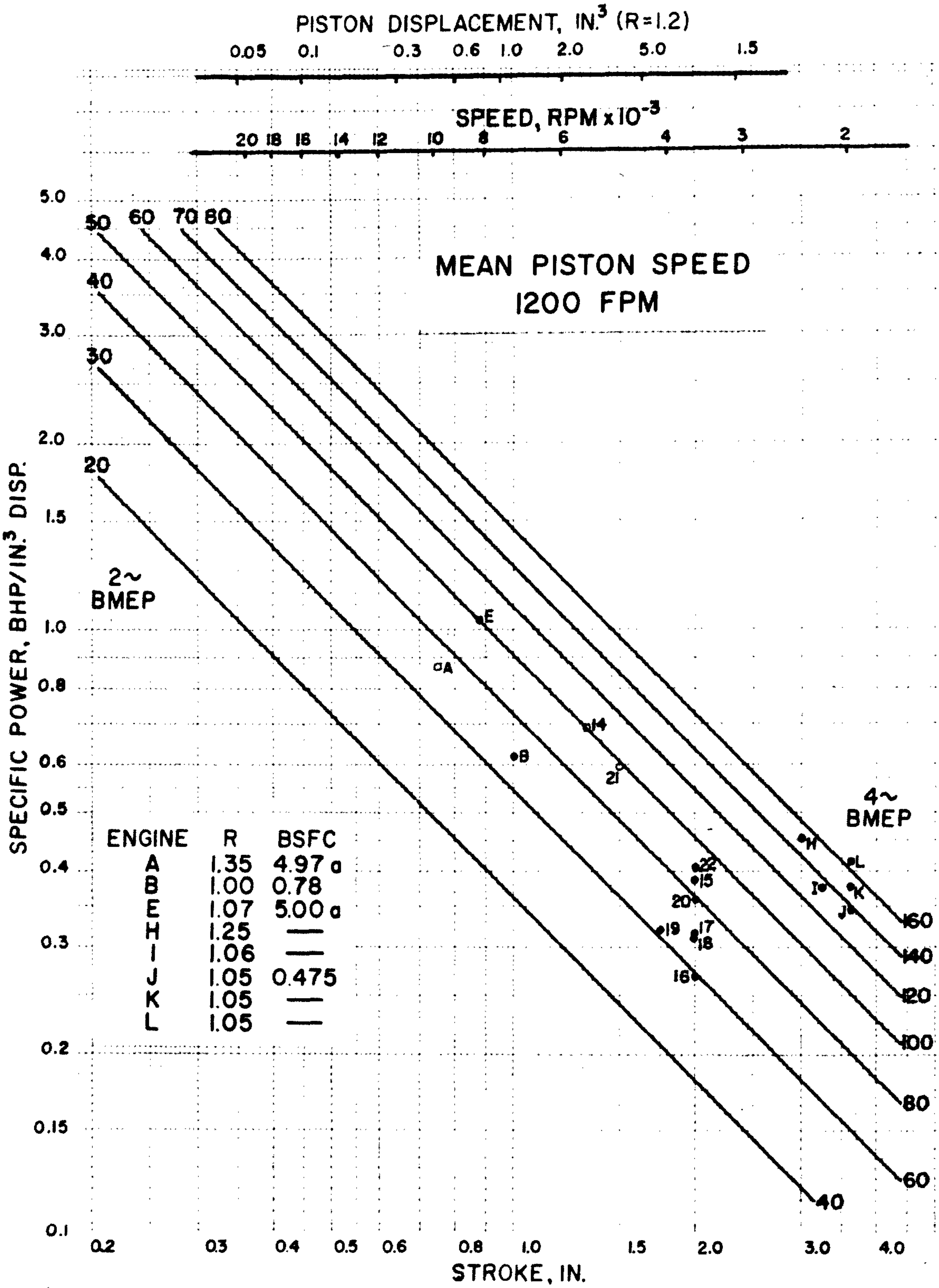


FIG. 7 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED

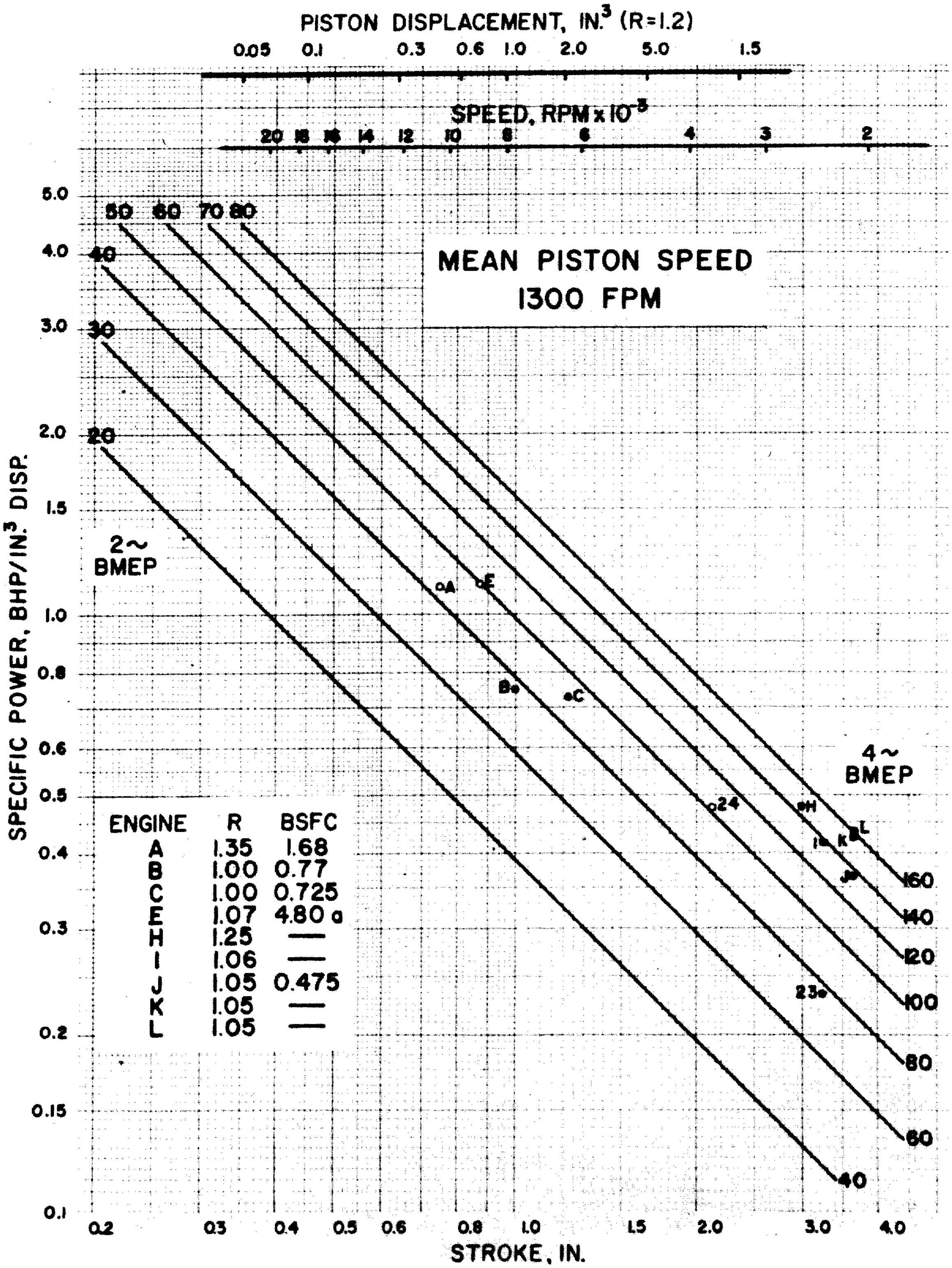
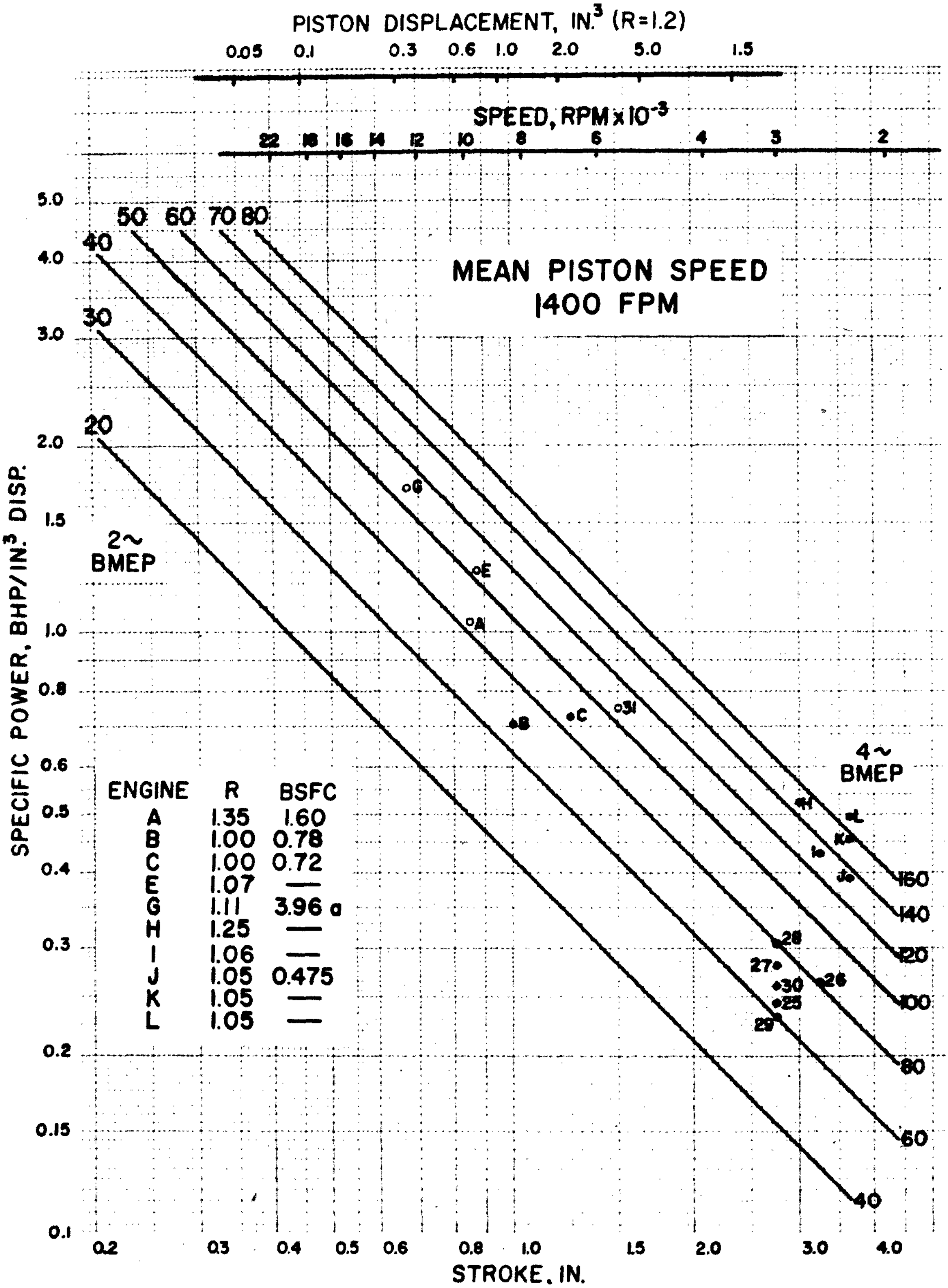


FIG.8 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED



ENGINE	R	BSFC
A	1.35	1.60
B	1.00	0.78
C	1.00	0.72
E	1.07	—
G	1.11	3.96 a
H	1.25	—
I	1.06	—
J	1.05	0.475
K	1.05	—
L	1.05	—

FIG.9 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED

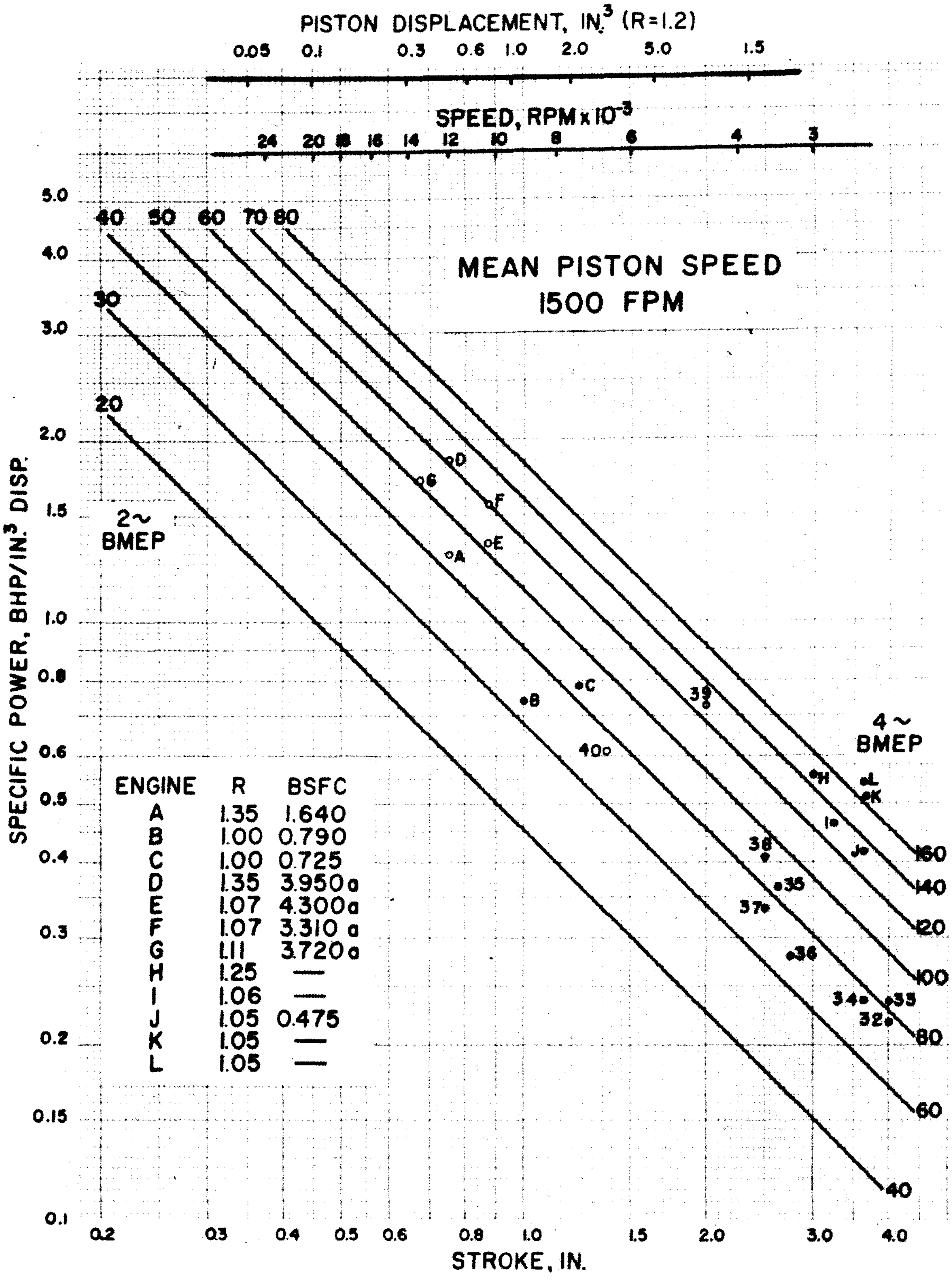


FIG.10 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED

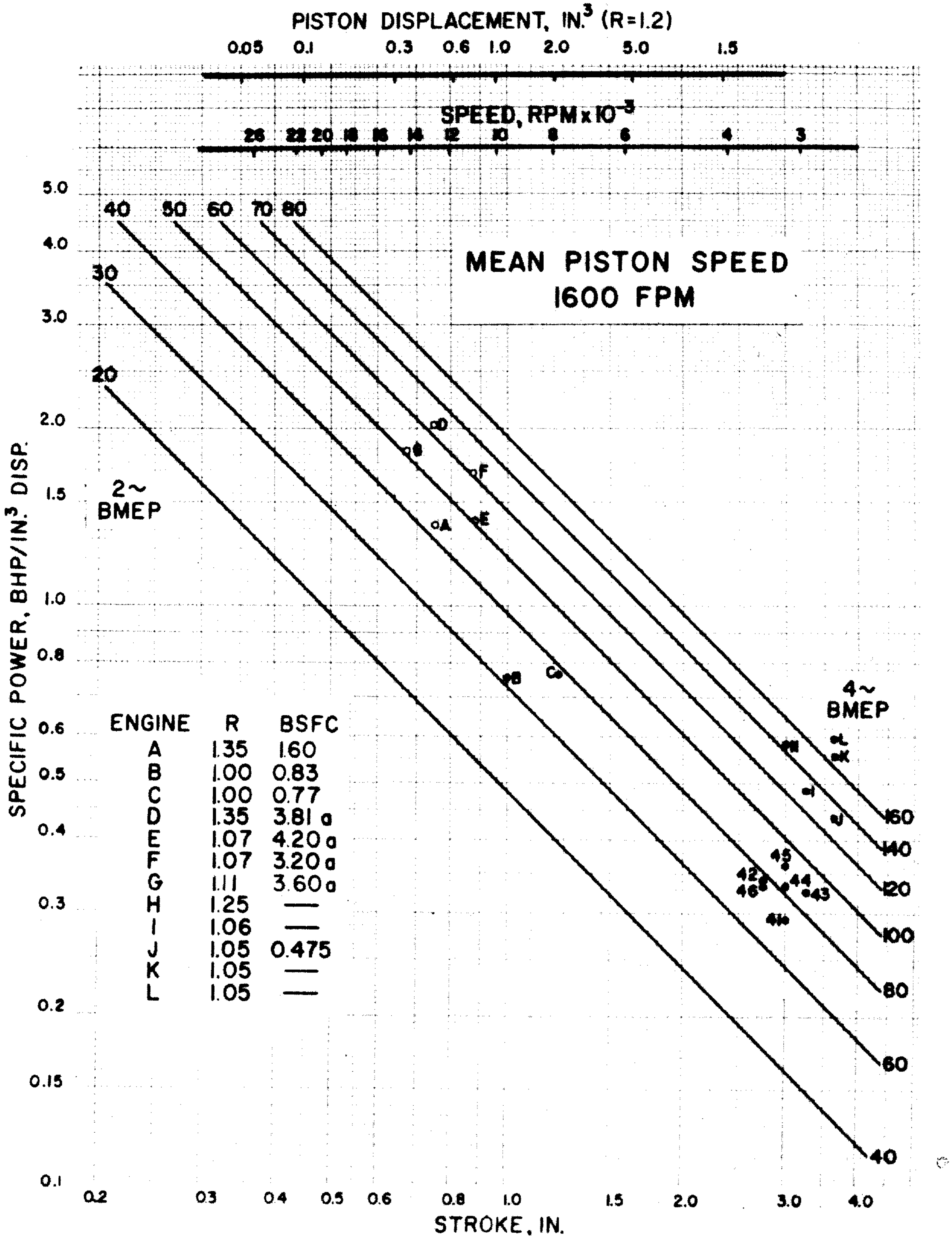


FIG. 11 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED

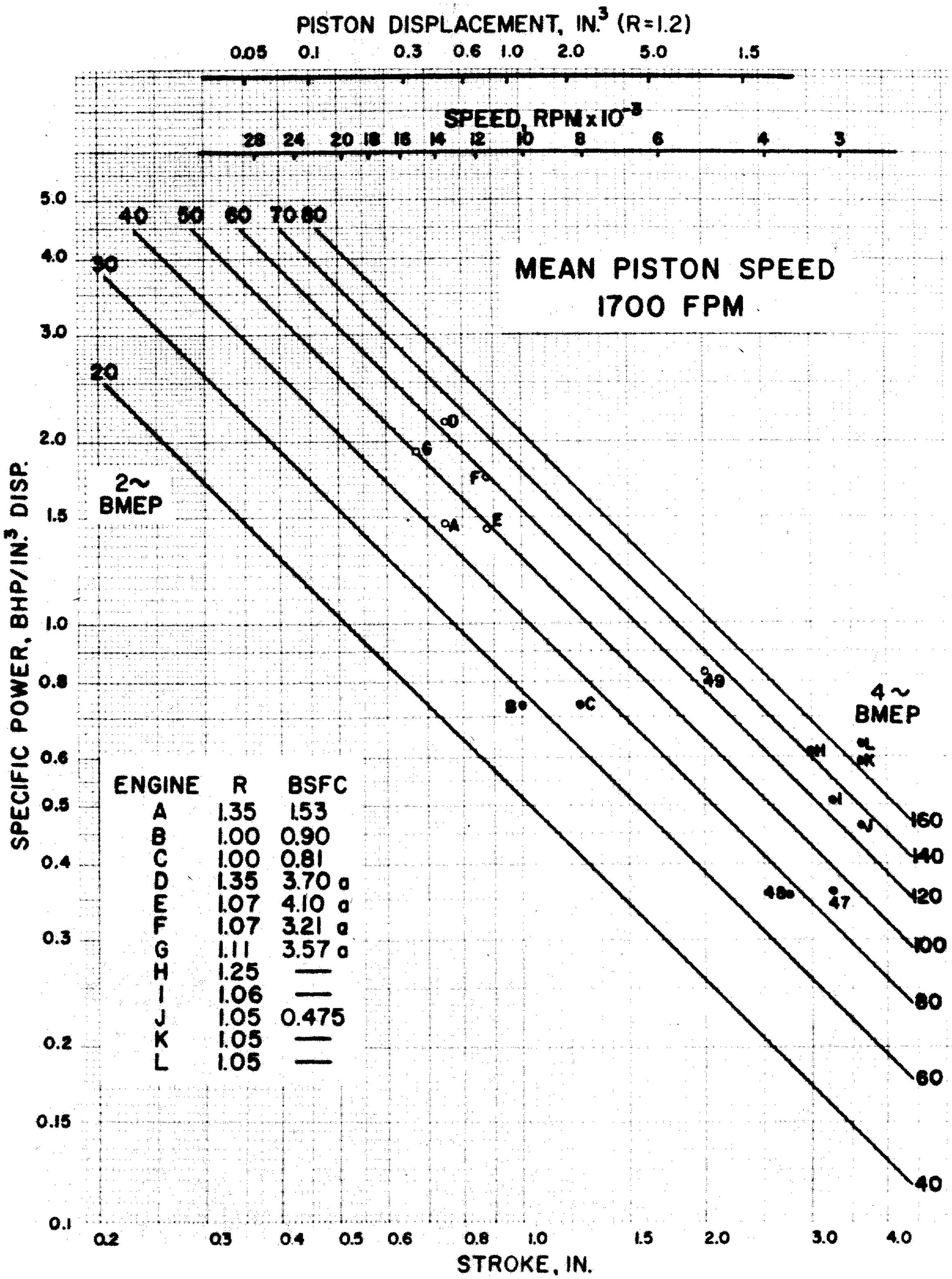


FIG.12 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED

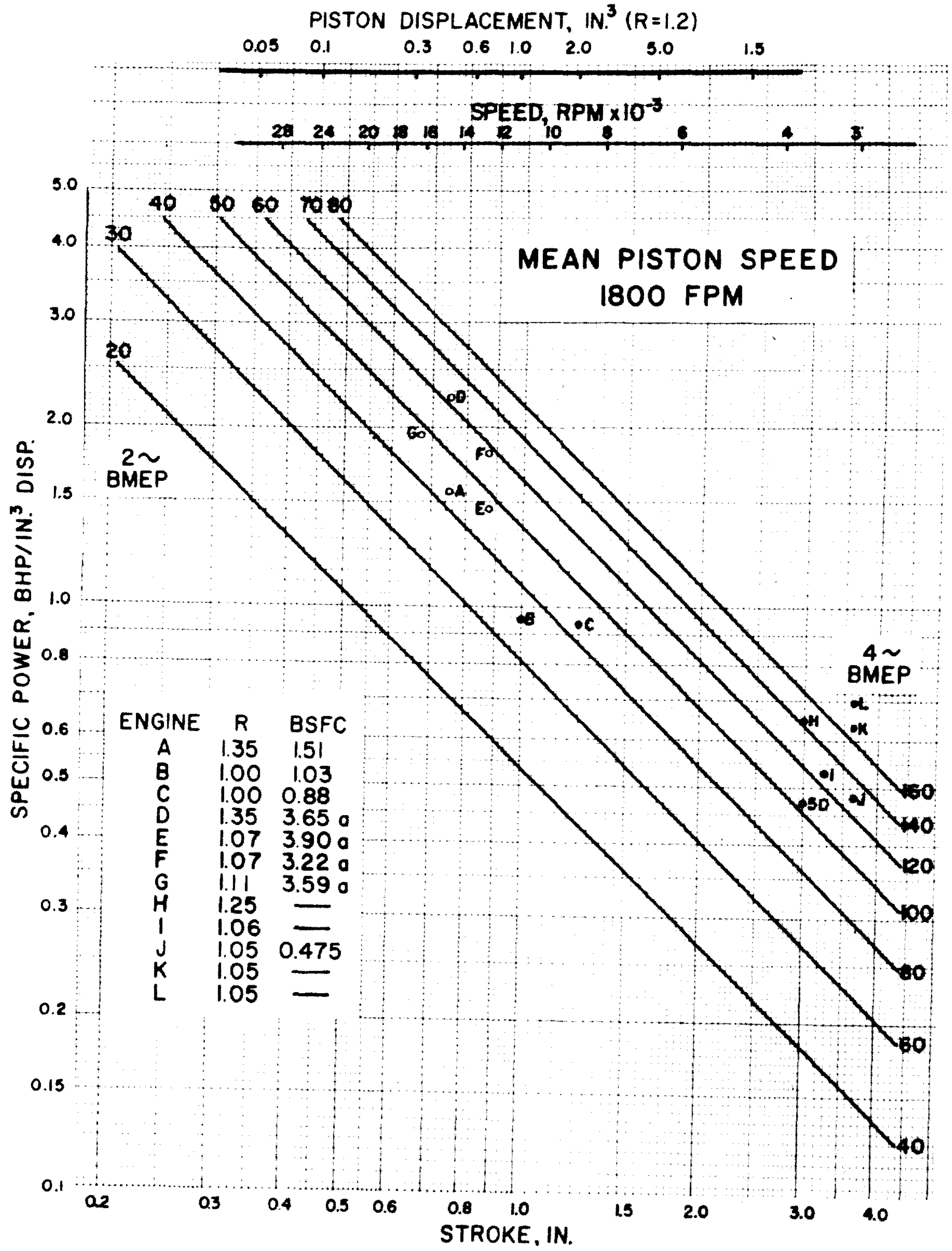


FIG. 13 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED

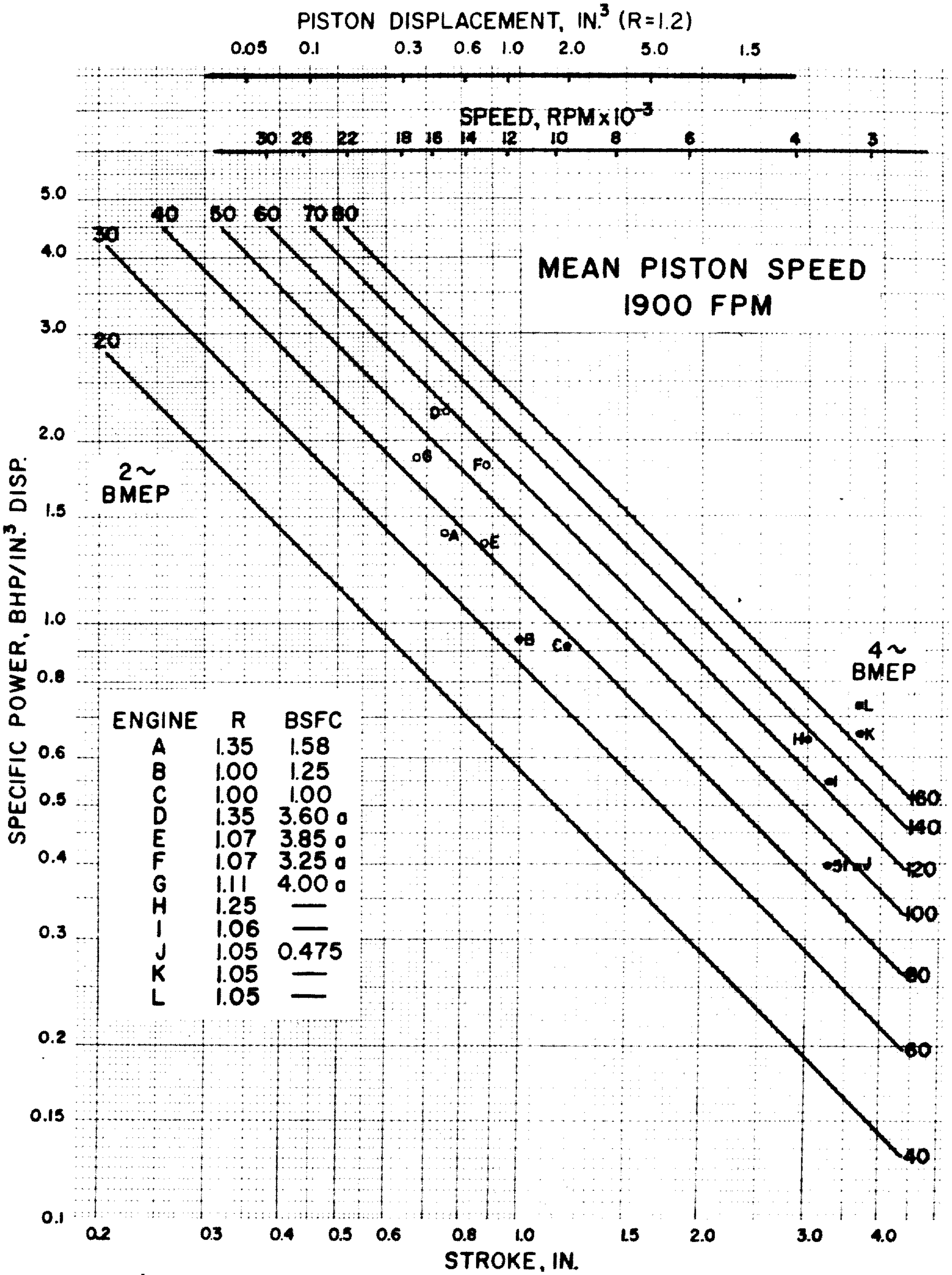
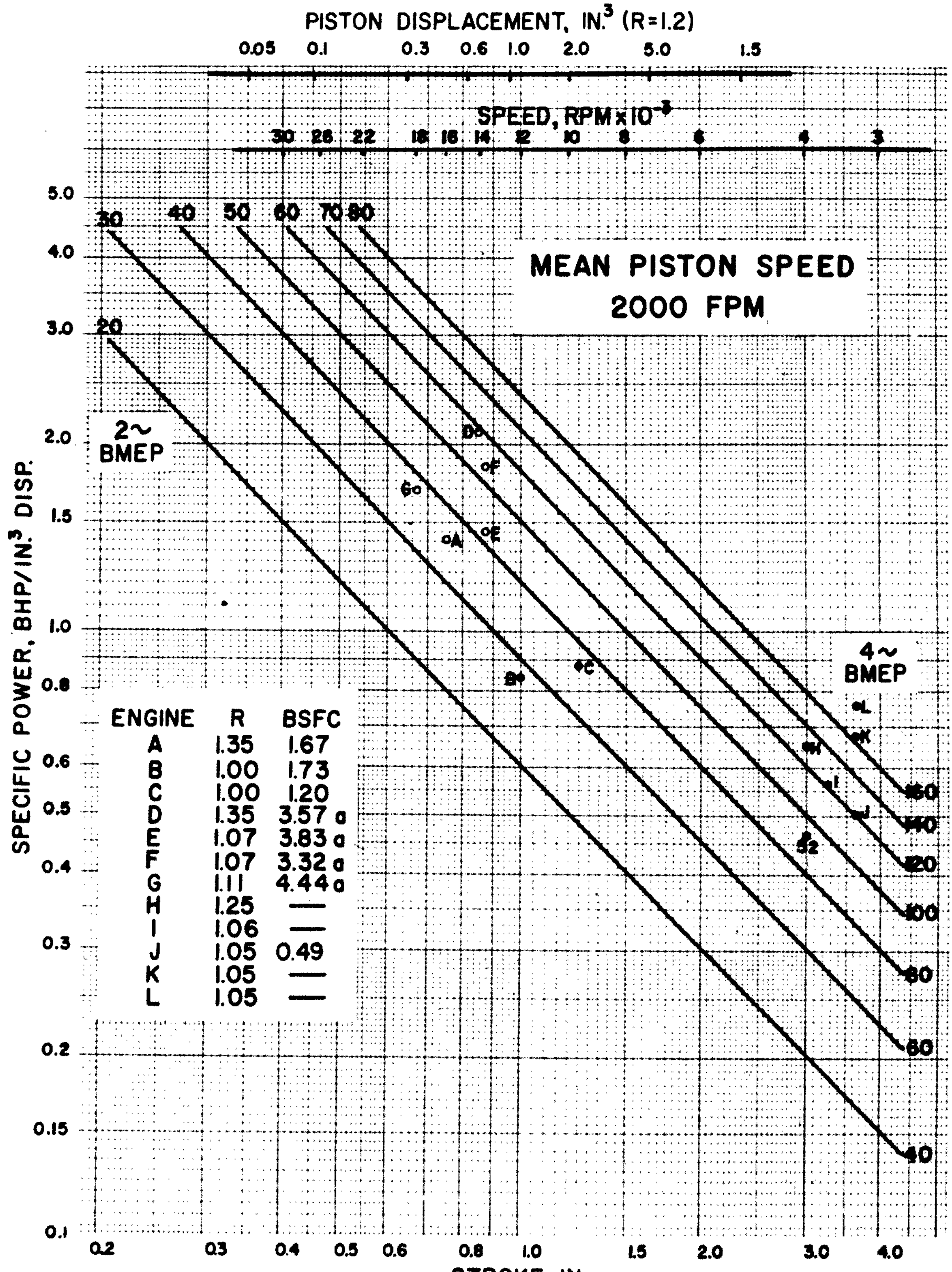


FIG.14 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED



ENGINE	R	BSFC
A	1.35	1.67
B	1.00	1.73
C	1.00	1.20
D	1.35	3.57 a
E	1.07	3.83 a
F	1.07	3.32 a
G	1.11	4.44 a
H	1.25	—
I	1.06	—
J	1.05	0.49
K	1.05	—
L	1.05	—

FIG.15 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED

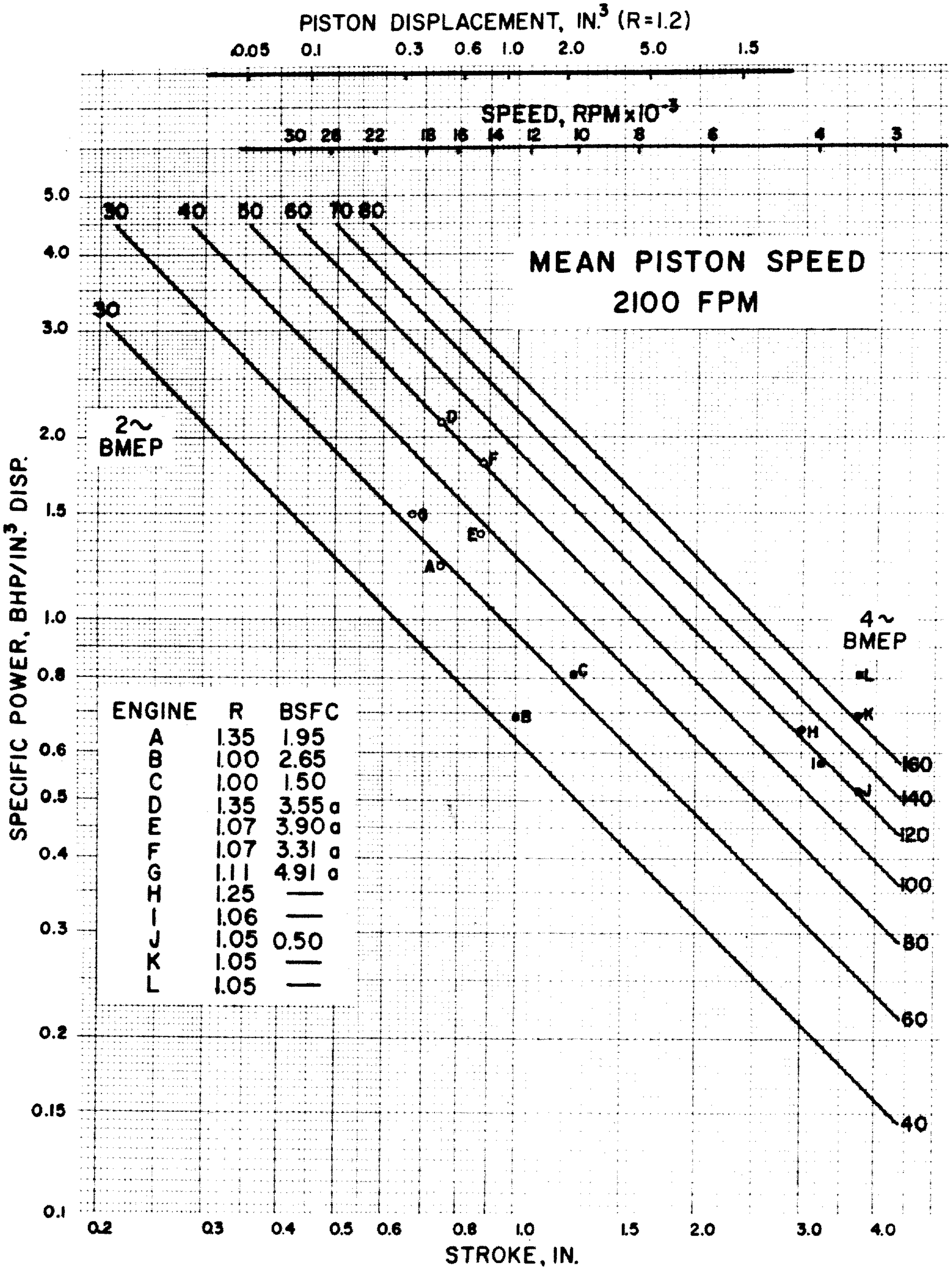


FIG. 16 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED

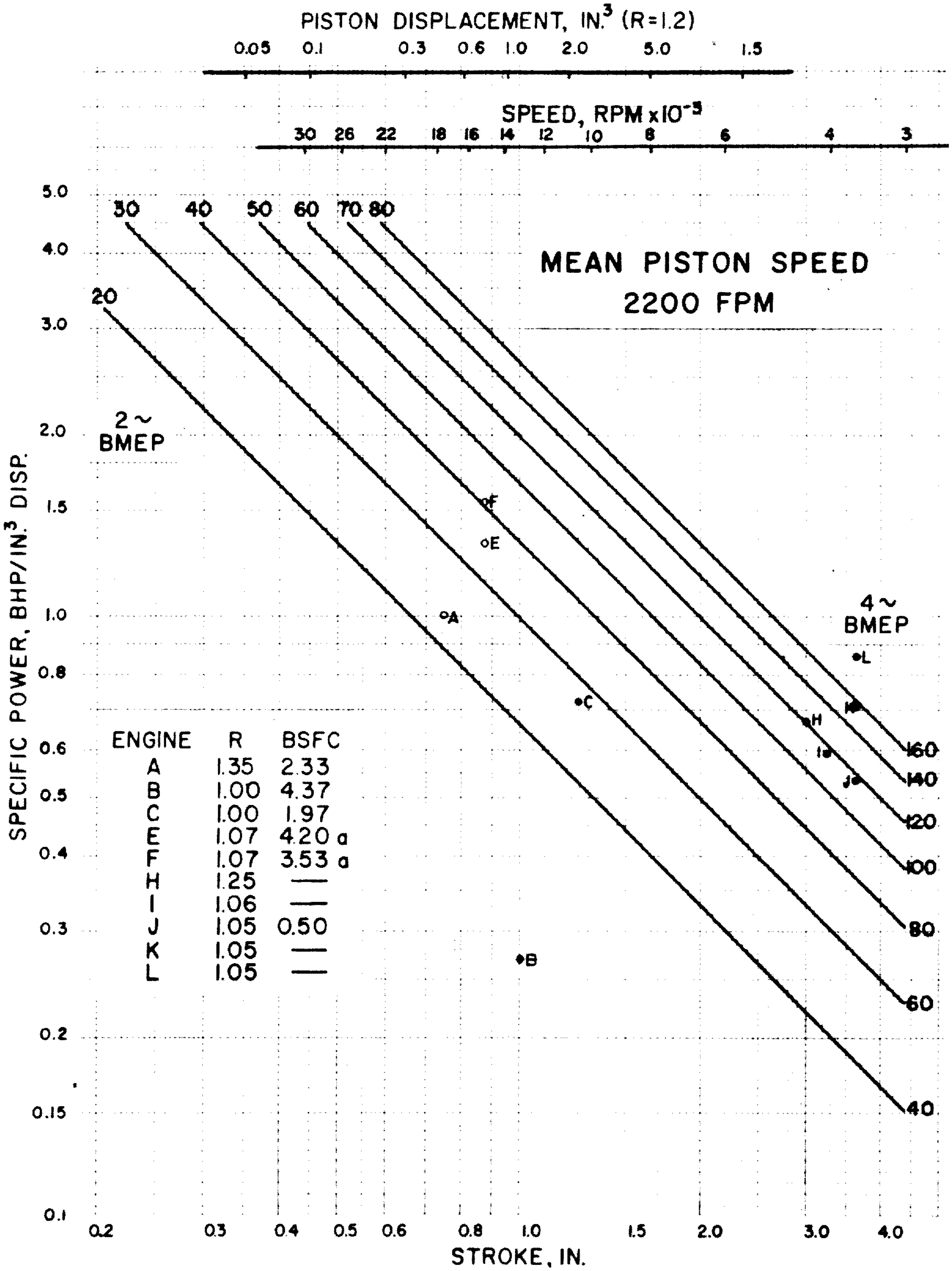


FIG.17 SPECIFIC POWER VS SIZE FOR VARIOUS VALUES OF BMEP AT CONSTANT PISTON SPEED

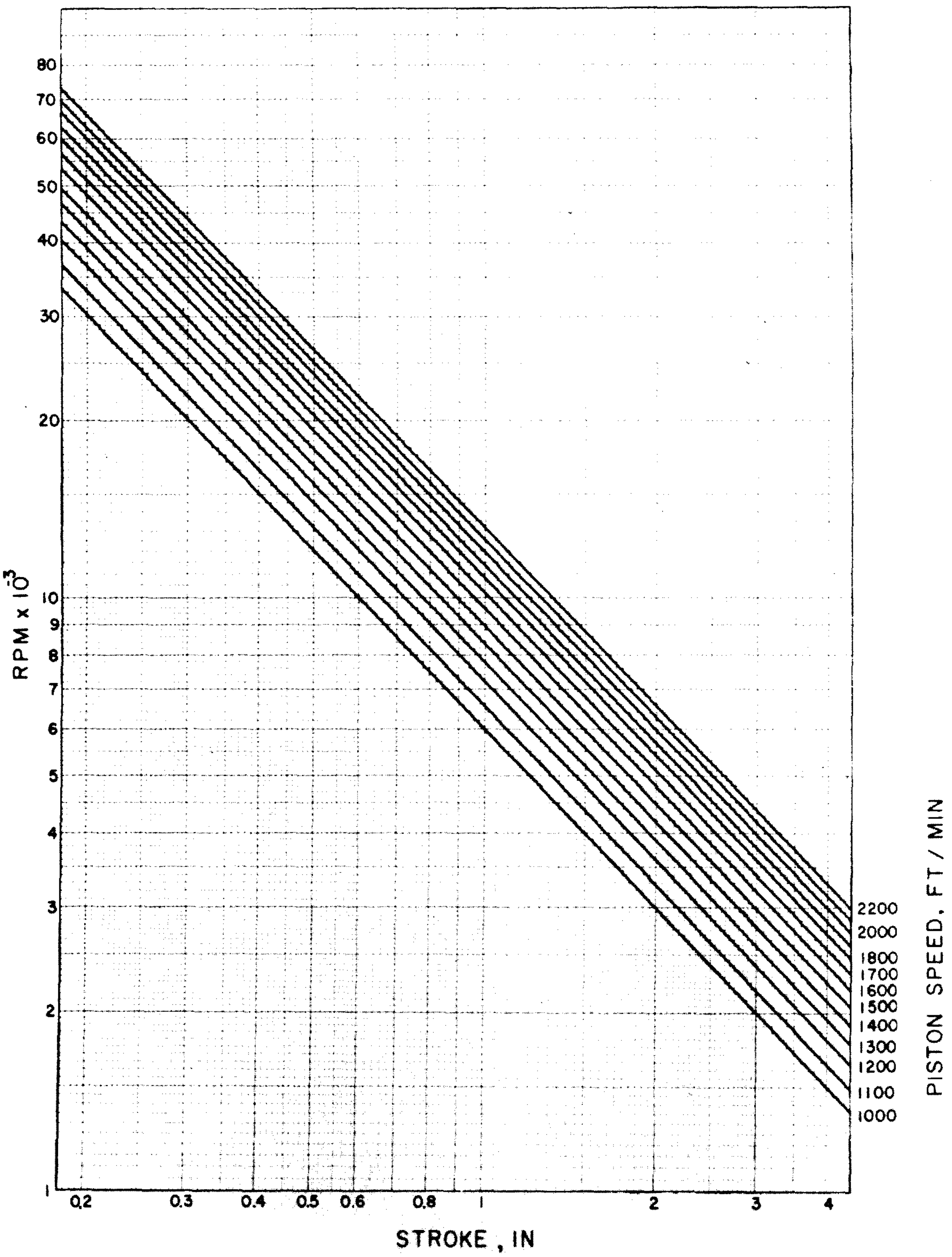


FIG. 18 RPM VS STROKE FOR VARIOUS PISTON SPEEDS

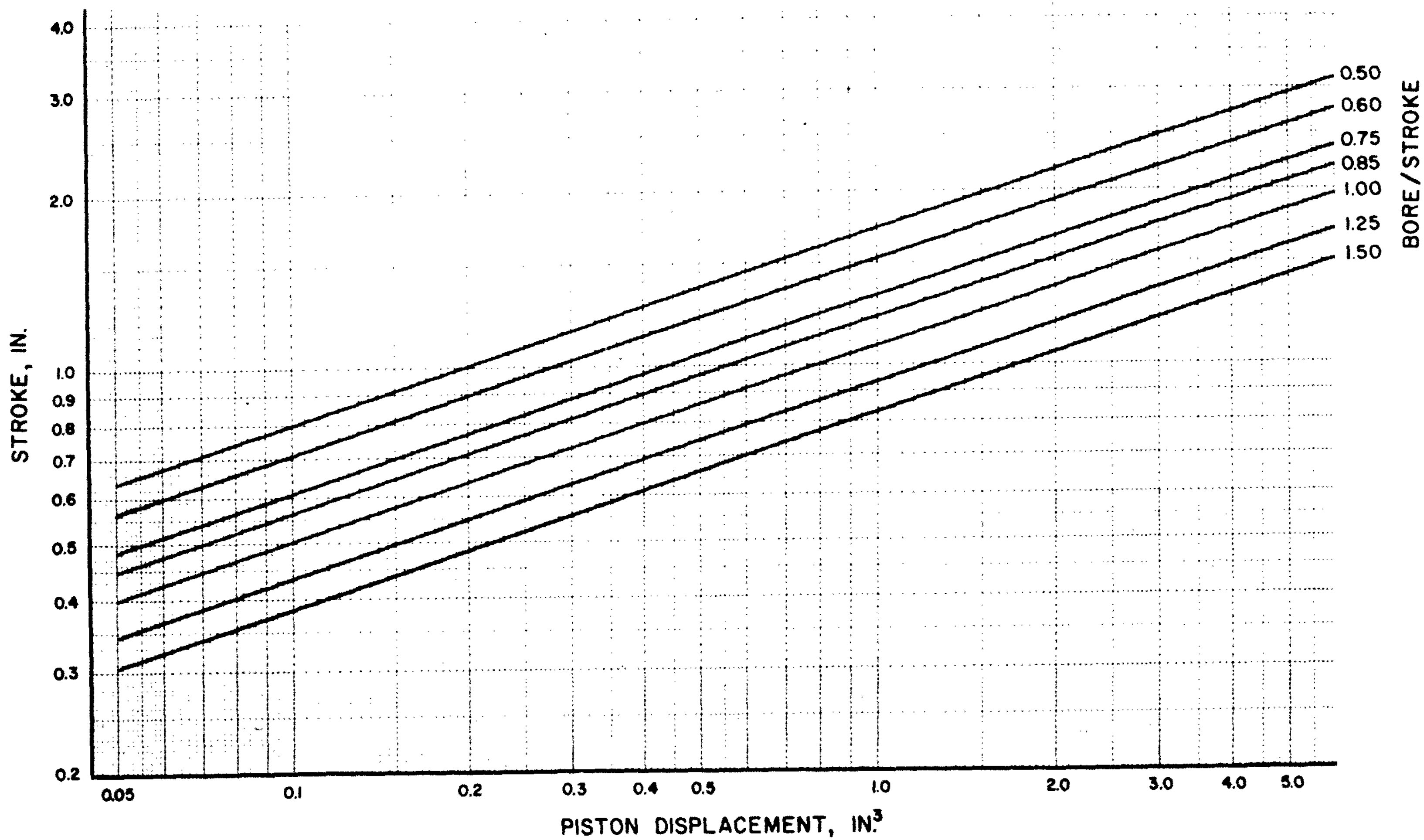


FIG. 19 STROKE VS PISTON DISPLACEMENT FOR VARIOUS BORE/STROKE RATIOS

TABLE 1

SPECIFICATIONS FOR ENGINES PLOTTED ON FIGS. 5 THROUGH 17

Engine	Mfr. & Model	Bore/stroke ratio, R	Stroke, inches	Cycle	Cooling	Engine Application
A	Test Engine-Cross Scav.	1.350	0.75	2	Air	Test
B	Test Engine-L-Head V	1.000	1.000	4	Air	Test
C	Test Engine-O.H.V.	1.000	1.250	4	Air	Test
D	Dooling 61	1.350	0.75	2	Air	MA
E	McCoy 60 "Red Head"	1.074	0.875	2	Air	MA
F	Hornet 60	1.072	0.875	2	Air	MA
G	McCoy 29 "Red Head"	1.113	0.670	2	Air	MA
H	Chevrolet V-8 "1955"	1.250	3.000	4	Liquid	Au
I	Dodge V-8 Truck	1.058	3.250	4	Liquid	Au
J	Chrysler V-8 "1951"	1.052	3.625	4	Liquid	Au
K	Chrysler V-8 Mod	1.052	3.625	4	Liquid	Au
L	Chrysler V-8 Mod High Comp.	1.052	3.625	4	Liquid	Au
1	Universal AFC	0.857	3.500	4	Liquid	Gn & In
2	Lauson 555-100	1.067	1.875	2	Air	LM
3	West Bend 2703	1.077	1.625	2	Air	LM
4	McCulloch 47	1.333	1.500	2	Air	CS
5	Onon 1-B	1.000	2.75	4	Air	Gn
6	Lauson 55 AB	1.067	1.875	4	Air	GP
7	Clinton 700	1.070	1.875	4	Air	GP
8	Onon AH	1.111	2.25	4	Air	Gn& In
9	Onon BH	1.111	2.25	4	Air	Gn& In
10	Power Prod. Av 80	1.125	2.00	2	Air	GP
11	Onon CW	1.143	3.50	4	Air	Gn
12	West Bend 2774	1.231	1.625	2	Air	CS
13	Clinton 800	1.270	1.875	4	Air	GP
14	McCulloch 33	1.273	1.325	2	Air	CS
15	Jacobsen 200	1.000	2.00	4	Air	LM
16	Briggs & Stratton 6	1.000	2.00	4	Air	GP
17	Continental AU 7B	1.060	2.00	4	Air	GP
18	Continental AU 8B	1.125	2.00	4	Air	GP
19	Reo 2980	1.143	1.75	4	Air	LM
20	Continental AU 85	1.156	2.00	4	Air	GP

21	Home Lite 26	1.158	1.500	2	Air	CS
22	Kohler 90	1.187	2.00	4	Air	GP
23	Wisconsin VF4	1.000	3.25	4	Air	GP
24	Home Lite 23	1.060	2.125	2	Air	Gn, P, & Br
25	Cushman M6	0.864	2.750	4	Air	GP
26	Wisconsin TF	1.000	3.250	4	Air	GP
27	Cushman M8	1.045	2.75	4	Air	GP
28	Lauson PAH	1.045	2.75	4	Air	GP
29	Onon LK	1.090	2.75	4	Air	Gn & In
30	Onon CK	1.091	2.75	4	Air	GP
31	McCulloch 4-30	1.412	1.50	2	Air	CS
32	Wisconsin AGH	0.875	4.00	4	Air	GP
33	Wisconsin VG4D	0.875	4.00	4	Air	GP
34	Le Roi 140	0.966	3.625	4	Liquid	GP
35	Briggs & Stratton 14	1.000	2.625	4	Air	GP
36	Kermuth "Sea Pop"	1.000	2.75	4	Liquid	M
37	Kermuth "Sea Twin"	1.100	2.50	4	Liquid	M
38	Kohler 160	1.150	2.50	4	Air	GP
39	McCulloch 7.55	1.250	2.00	2	Air	CS & ED
40	McCulloch 33	1.273	1.375	2	Air	CS
41	Gladden 40	0.834	3.00	4	Air	GP
42	Wisconsin ABN	0.909	2.75	4	Air	GP
43	Wisconsin AEN	0.923	3.25	4	Air	GP
44	Gladden 75-ES	0.960	3.00	4	Air	GP
45	Gladden 75	0.960	3.00	4	Air	GP
46	Wisconsin AKN	1.045	2.75	4	Air	GP
47	Clinton 2500	0.960	3.25	4	Air	GP
48	Onon ACK	1.091	2.75	4	Air	Gn
49	McCulloch 99	1.250	2.00	2	Air	CS, P & ED
50	Home Lite 32	1.000	3.00	2	Air	Gn
51	Kohler 660	1.115	3.25	4	Air	GP
52	Gladden MC	0.960	3.00	4	Air	MC

REFERENCES:

Engines A-G - Project Test Data
" H Ref. 24
" I Ref. 25
" J-L Ref. 20
" 1-52 Ref. 26

ENGINE APPLICATION:

Au	Automotive	In	Industrial
Br	Blower	LM	Lawn Mower
CS	Chain Saw	M	Marine
ED	Earth Drill	MA	Model Airplane
Gn	Generator	MC	Motor Cycle
GP	General Purpose	P	Pump

performance points for new engines as the data become available, since increasing activity in the miniature engine field will undoubtedly take place in years to come.

An example of a method for use of these curves for preliminary engine design follows.

Example:

Assume the problem is to determine the cylinder size of a simple two-cycle engine of the following specification:

a. Maximum power at sea level, 0.75 hp
b. Shaft speed, 12,000 rpm
c. Mean piston speed, 1,500 fpm

Referring to Figure 18 or to Figure 10, the required length of stroke is found to be 0.75 inches. Figure 10 can now be used to estimate a BMEP by comparison with current practice in this size range. It is obvious that a BMEP of 40 psi is attainable and in fact quite conservative. Assuming that only a limited development time can be spent on this engine, a design BMEP of 40 psi is chosen and the specific power capacity is found to be 1.2 hp per cubic inch of displacement.

For an output of 0.75 hp, the displacement must be 0.75/1.2 equals 0.625 cubic inch. However, the bore/stroke ratio (R) from Figure 19 is found to be 1.375, a value somewhat higher than those found in current practice.

This bore/stroke ratio can be reduced by increasing the design BMEP or by using a higher mean piston speed. By increasing the mean piston speed to 1600 fpm, the bore/stroke ratio is found to be a more conservative 1.25, but the resulting engine with higher piston speed would have a somewhat shorter life expectancy. A reduction to the same bore/stroke ratio could also be effected by increasing the design BMEP to 48 psi; however, in this case, development time might be increased appreciably.

It is obvious that many combinations of engine rotating speed, mean piston speed, and horsepower will be infeasible in terms of compatibilities established by current practice. Some method of trial-and-error procedure such as the one described above will be necessary to determine the most practical design for each particular set of requirements.

2.1.2 Miniature Test-Engine Performance

Considering the broad scope of the project, it was necessary to confine the experimental phase of the work to test evaluation of a few typical engines representative of the more common configurations. These

FIG.20 ASSEMBLED AND DISASSEMBLED VIEWS OF CROSS-SCAVENGED TWO-CYCLE TEST ENGINE

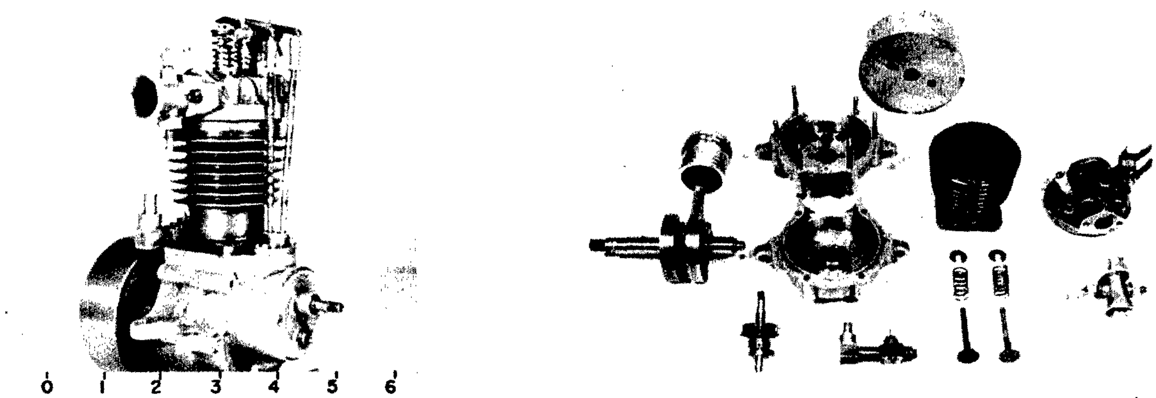


FIG.21 ASSEMBLED AND DISASSEMBLED VIEWS OF OVERHEAD-VALVE FOUR-CYCLE TEST ENGINE

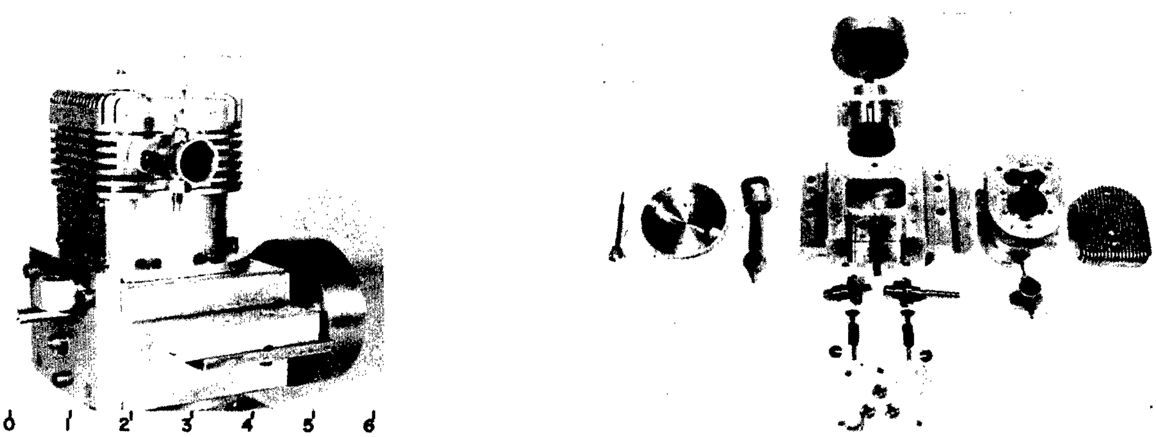


FIG.22 ASSEMBLED AND DISASSEMBLED VIEWS OF L-HEAD FOUR-CYCLE TEST ENGINE

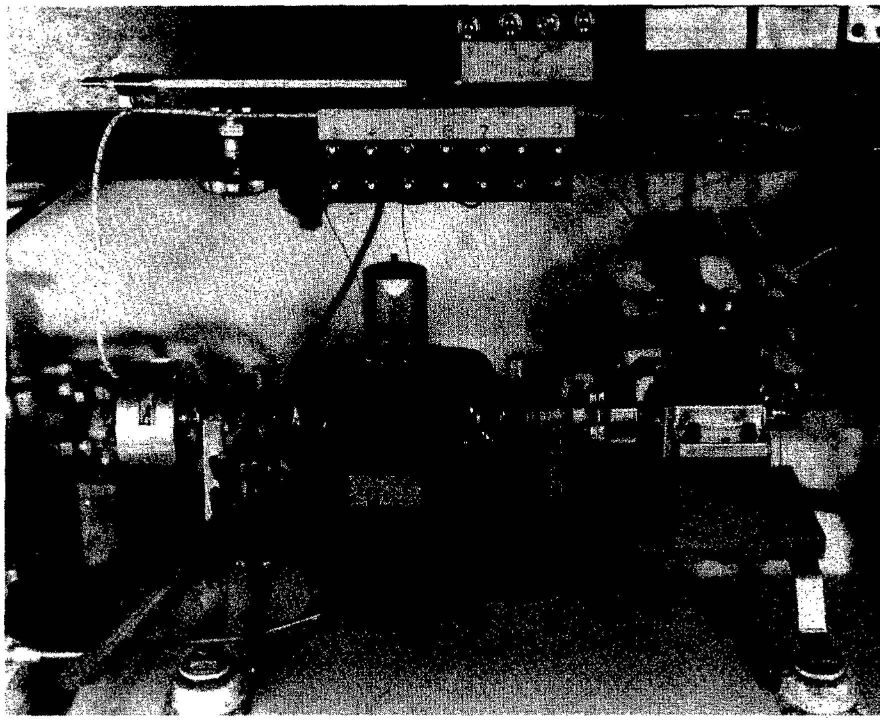


Figure No. 23. Miniature Engine Test Dynamometer With Two-Cycle Test Engine Installed..

included loop- and cross-scavenged, crankcase-compression two - cycle engines and overhead-valve and L-head four-cycle engines. The test work was designed to explore the power, economy, and endurance potentials of the engines under a variety of operating conditions with various combinations of performance parameters such as compression ratio, valve timing, operating temperature, etc.

Figures 20 through 22 show views of the test engines, assembled and disassembled. It is obvious that the engines were designed for the ruggedness necessary for continual dynamometer test-work as well as for convenience of independent parameter variation. Production design of these engines could evolve considerably lighter and less bulky units. Figure 23 shows the two-cycle test engine installed on the engine test dynamometer. A discussion of the test facilities suitable for test and development work with miniature engines and generators is presented in WADC TR 53-180 (Ref. 27).

Typical performance curves for the project test engines are shown in Figures 24 through 27. Complete design and performance specifications for the engines associated with these particular curves are given in Table 2. It will be noted that these are all air-cooled engines operating on 80-octane aviation gasoline fuel. Close comparison of these particular curves is not justified, since the engines were not necessarily developed to the same degree. An important characteristic to be noted throughout all the project test work, however, is the lower brake specific fuel consumption associated with the four-cycle engines.

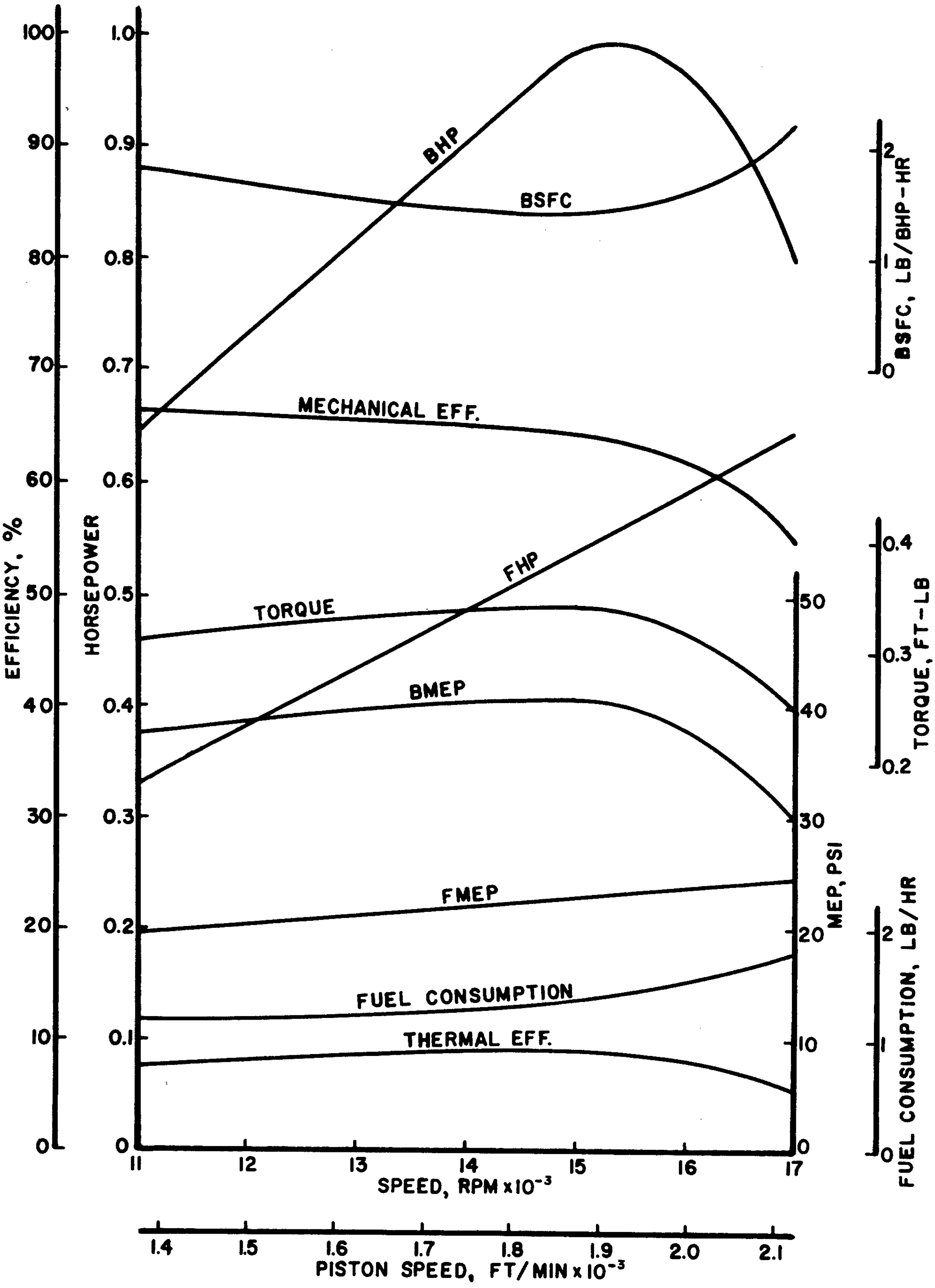


Figure No. 24. Typical Performance Curves for Cross-Scavenged Two-Cycle Test Engine.

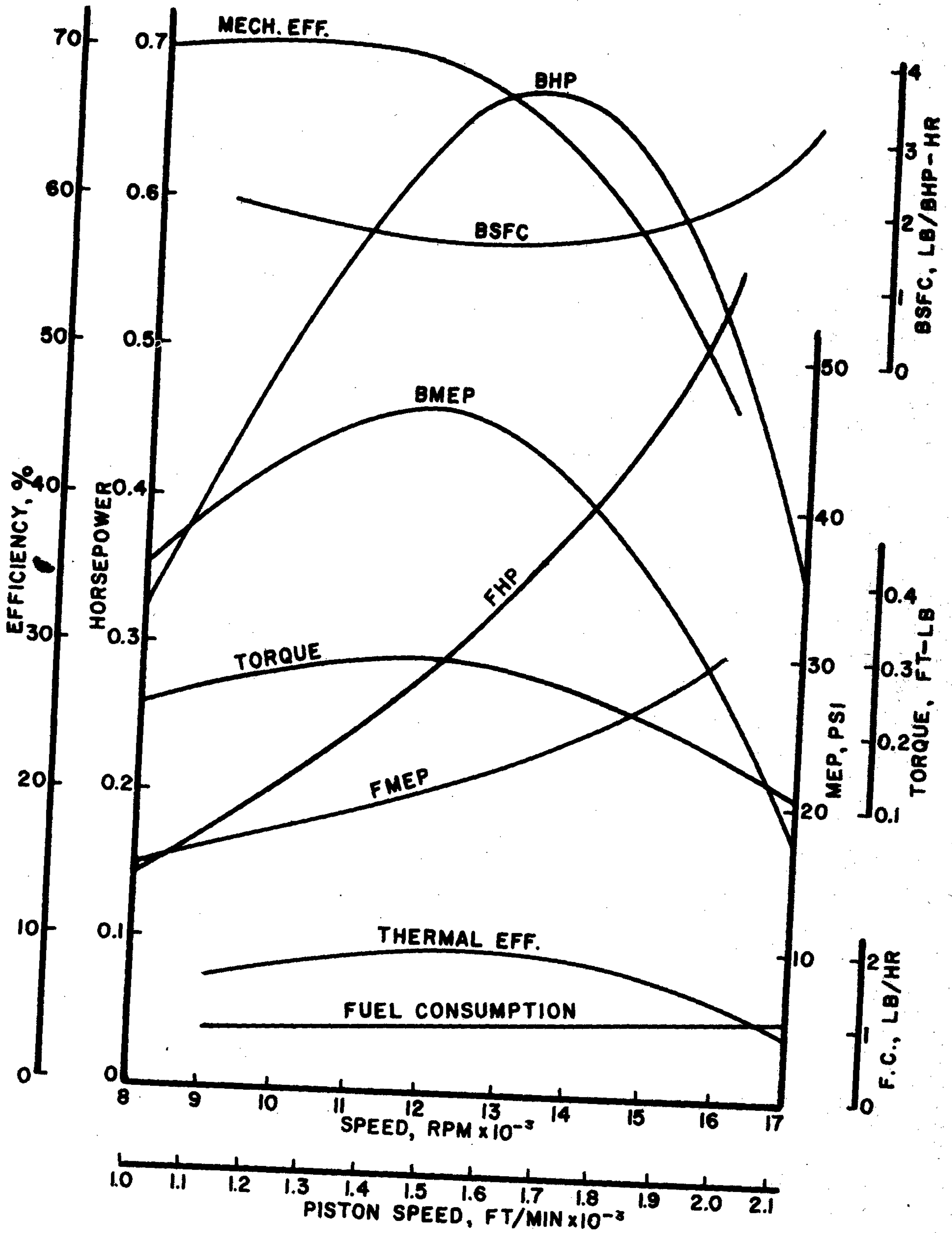


Figure No. 25. Typical Performance Curves for Loop-Scavenged Two-Cycle Test Engine.

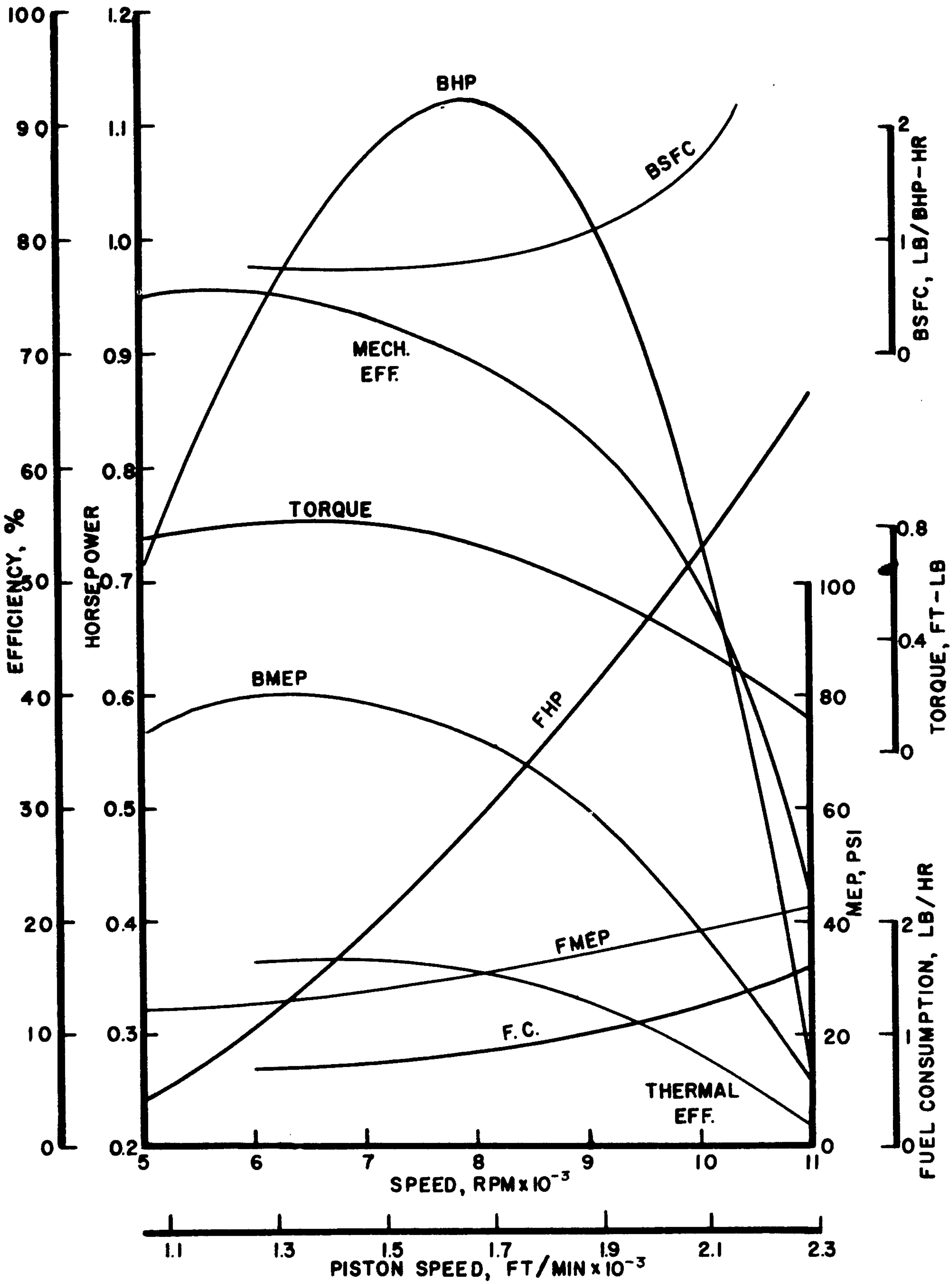


Figure No. 26. Typical Performance Curves for Overhead-Valve Four-Cycle Test Engine.

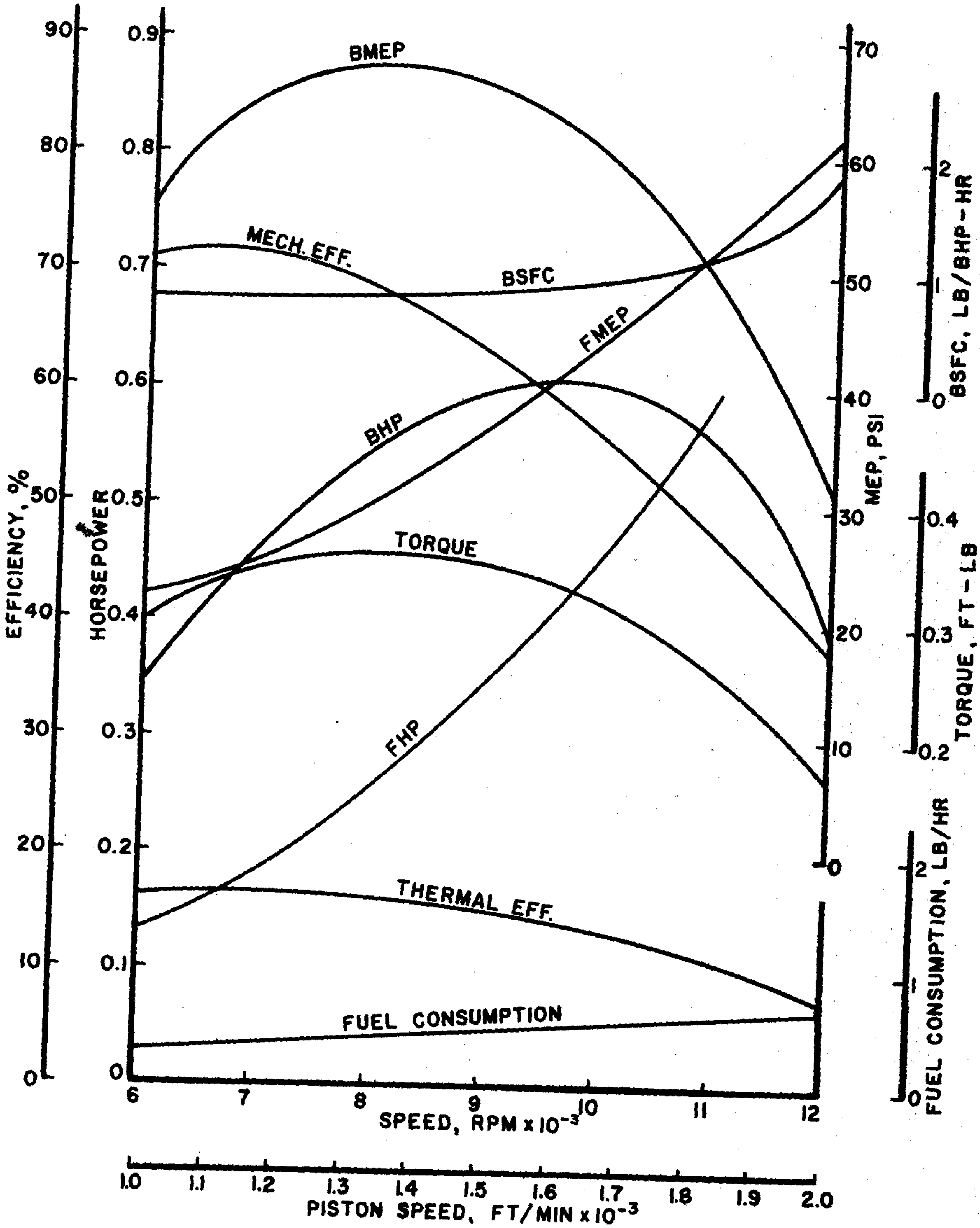


Figure No. 27. Typical Performance Curves for L-Head Four-Cycle Test Engine.

TABLE 2

TEST-ENGINE PARAMETERS USED FOR TESTS

DESCRIBED IN FIGURES 24 THROUGH 27

		Report figure number			
		24	25	26	27
Cycle		2	2	4	4
Type		Cross-Scav.	Loop-Scav.	Ovhd.-valve	L-head
Bore, in.		1.015	0.890	1.250	1.015
Stroke, in.		0.750	0.750	1.250	1.000
Displacement, cu. in.		0.607	0.466	1.534	0.809
Bore/stroke		1.350	1.187	1.000	1.015
Rod length/Crank radius (l/r)		3.667	3.667	4.000	5.17
Rod bearing type		Roller	Roller	Plain	Plain
Main bearing type		Ball	Ball	Plain	Plain
Crankpin, dia/bore		0.255	0.291	0.340	0.308
Crankpin, proj. area/bore area		0.080	0.104	0.151	0.168
Main brg., dia/bore		0.493	0.562	0.350	0.466
Exh. port-area/bore-area		0.333	0.066	0.122	0.205
Inl. port-area/bore area		0.284	0.139	0.122	0.205
Piston, length/bore		0.862	0.898	1.000	1.170
Ext. timing, deg.	open	73-BBC	72-BBC	60-BBC	52-BBC
	close	73-ABC	72-ABC	16-ATC	12-ATC
Int. timing deg.	open	54-BBC	57.5-BBC	16-BTC	14-BTC
	close	54-ABC	57.5-ABC	56-ABC	55-ABC
Crankcase timing, deg.	open	122-BTC	122-BTC	N. A.	N. A.
	close	58-ATC	58-ATC		
Fuel		80-Octane Aviation Gasoline			
Lubricant		SAE #70 10% in fuel	SAE #70 10% in fuel	SAE #20 in crankcase	SAE #20 in crankcase
Compression ratio		6.9:1	16.3:1	6:1	6.3:1
Ignition timing, deg. BTC		53	53	48	29
Cylinder head temp, °F.		350	300	300	300
Crankcase comp. ratio		1.277	1.263	N. A.	N. A.
Carburetion		Needle-valve, 7/16 in. dia venturi			

N. A. - not applicable

To afford a clear and concise presentation of the performance attained to date with the project test engines, the test results have been consolidated into envelope curves showing BMEP, FMEP, and BSFC plotted versus engine speed. Figures 28 through 31 show the results with the three test engines on which sufficient data were accumulated to justify such a presentation, the cross-scavenged two-cycle, overhead-valve four-cycle, and L-head four-cycle engines. Figures 28 and 29 show results with the cross-scavenged two-cycle engine with alcohol-base fuel and petroleum-base fuel respectively. Figures 30 and 31 show the results with the two four-cycle engines on petroleum-base fuel. These envelope curves represent the best results obtained with the test engines during all of the project test work, i.e., the maximum BMEP and the minimum FMEP and BSFC.

Examination of Figures 28 and 29 shows an interesting comparison of alcohol and petroleum fuels used in the same engine, notably the difference between the BSFC curves for the two fuels. The minimum BSFC with alcohol fuel was 3.42 lb/bhp-hour at 14,300 rpm (1,785 fpm piston speed), and with petroleum fuel was 1.45 lb/bhp-hour at 14,200 (1,770 fpm piston speed). For miniature engine-generator set applications, petroleum fuels would appear to be preferable for all applications with operating time in excess of a few minutes, since the weight of petroleum fuel required would be considerably less than the weight of alcohol fuel for the same operating time. It is of interest that the maximum brake thermal efficiencies corresponding to the minimum BSFC points are equivalent and equal to 10.5 per cent, and that the speed of best economy is approximately the same with either fuel.

The maximum torques produced by the two fuels occur at approximately the same speed and are also very nearly equal; BMEP of 42 psi at 12,800 rpm and 42.6 psi at 13,200 rpm for alcohol and petroleum fuels respectively. The torque and power obtained with alcohol fuel would appear to be slightly better than those with petroleum fuel at speeds of 12,000 rpm and higher. The measurement of the friction torque of the two-cycle test engines is only approximate at best, since fuel flow must be continued to provide lubrication and as a consequence the engine temperatures can not be maintained equal to the operating temperatures. The FMEP curves obtained with the two fuels are thus considered comparable within the accuracy of measurement.

From Figures 30 and 31 it is apparent that the economy of the four-cycle test engines is considerably better than that of the two-cycle test engine. The minimum BSFC values were 0.66 and 0.79 lb/bhp-hour for the overhead-valve and L-head engines respectively with corresponding brake thermal efficiencies of 20.3 and 17 per cent. The maximum economy of the four-cycle test engines occurred at approximately the same speed as that of maximum torque (BMEP), and for both engines it is found at piston speeds between 1200 and 1300 ft./min. Moreover, the BSFC curves for the four-cycle engines are relatively "flat" over the range of 1000 to 1600 ft./min piston speed.

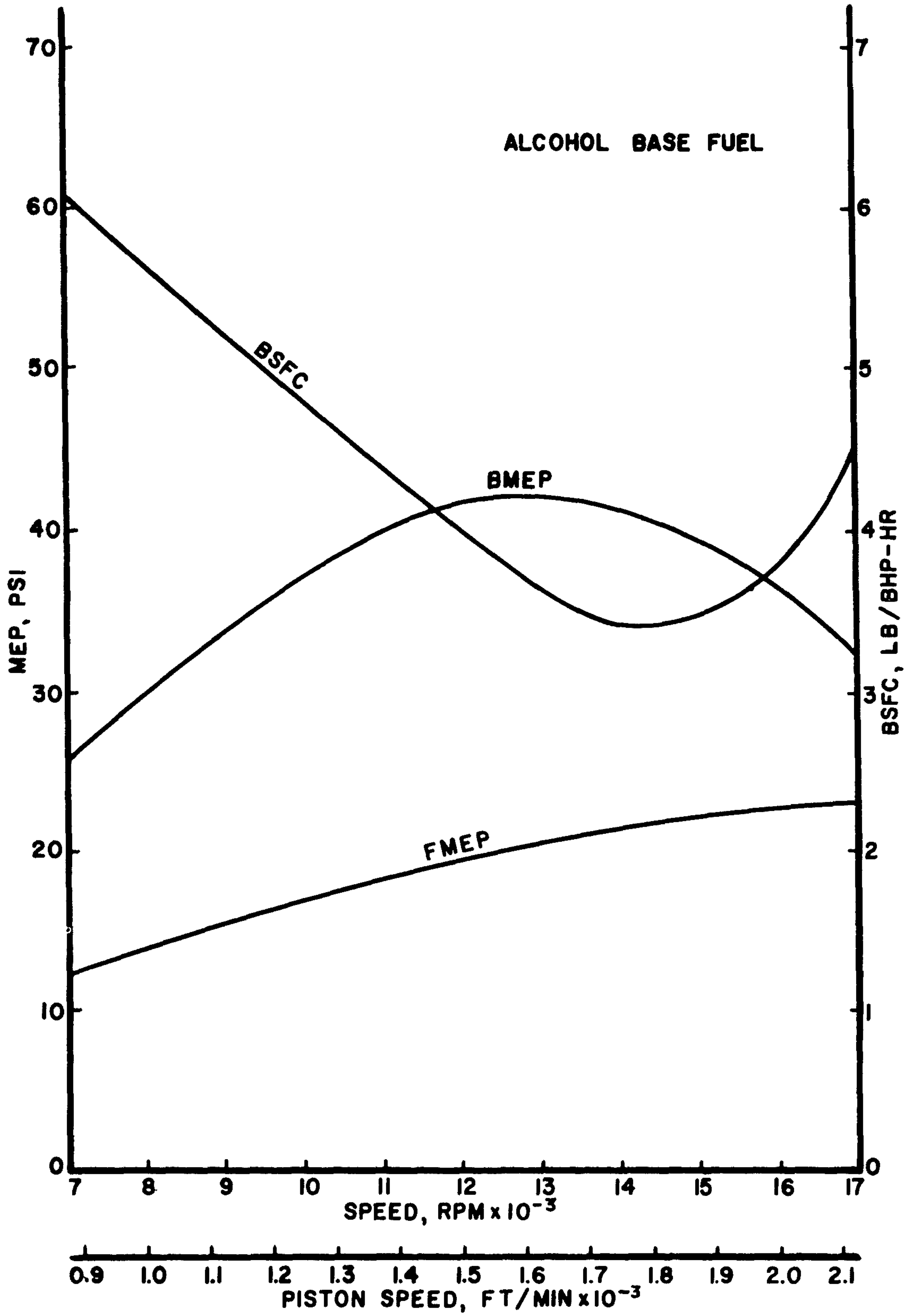


Figure No. 28. Envelope Curves for Performance of Cross-Scavenged Two-Cycle Test Engine.

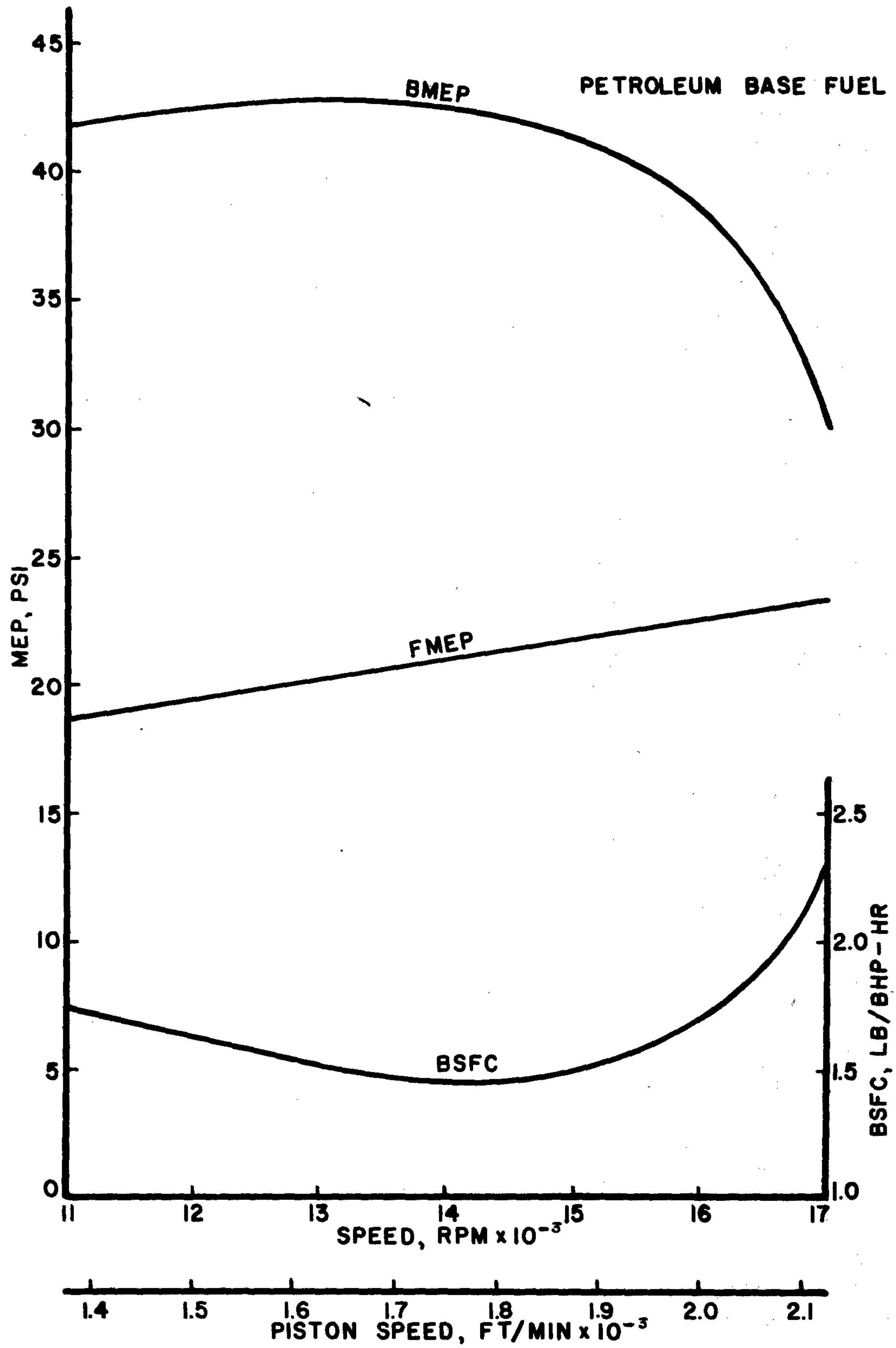


Figure No. 29. Envelope Curves for Performance of Cross-Scavenged Two-Cycle Test Engine.

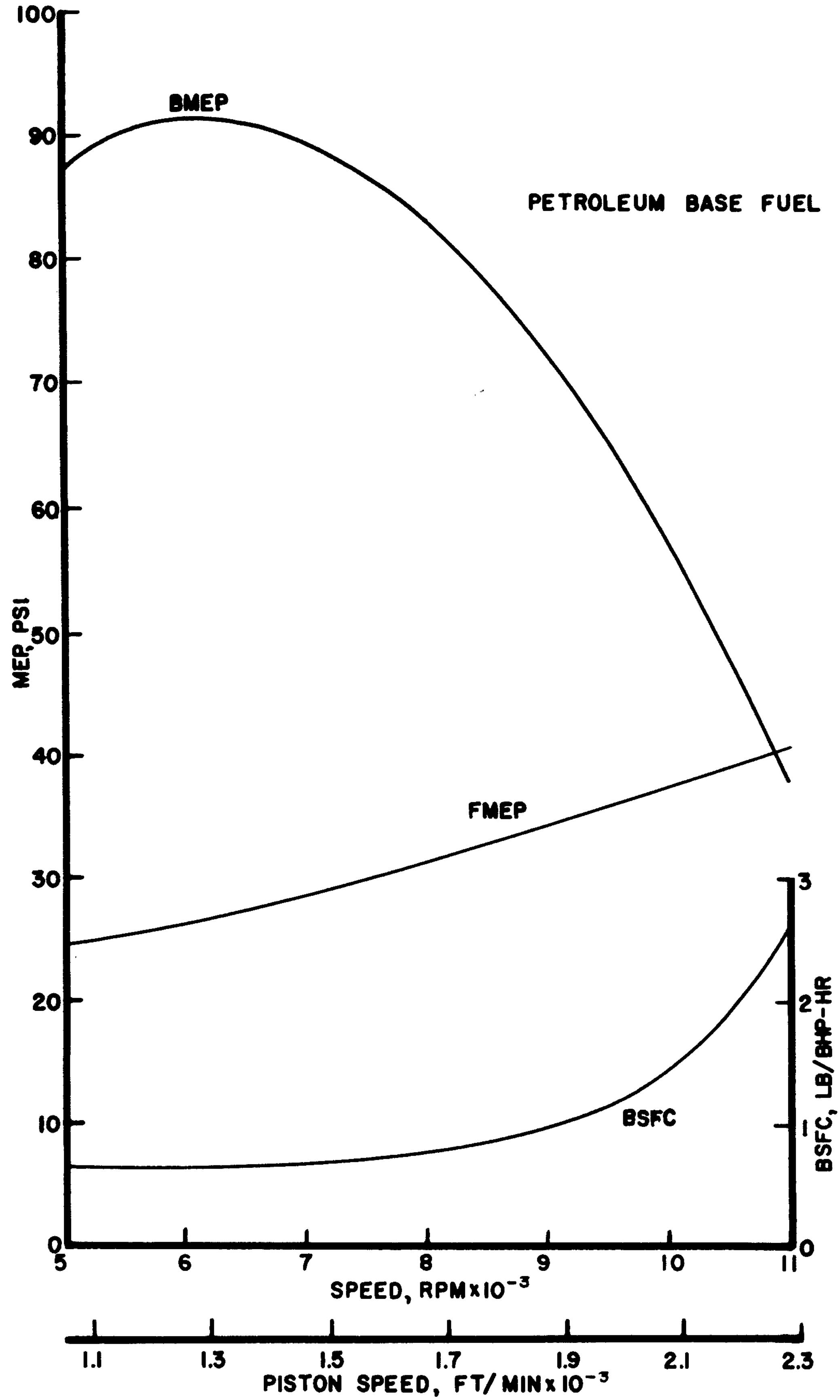


Figure No. 30. Envelope Curves for Performance of Overhead-Valve Four-Cycle Test Engine.

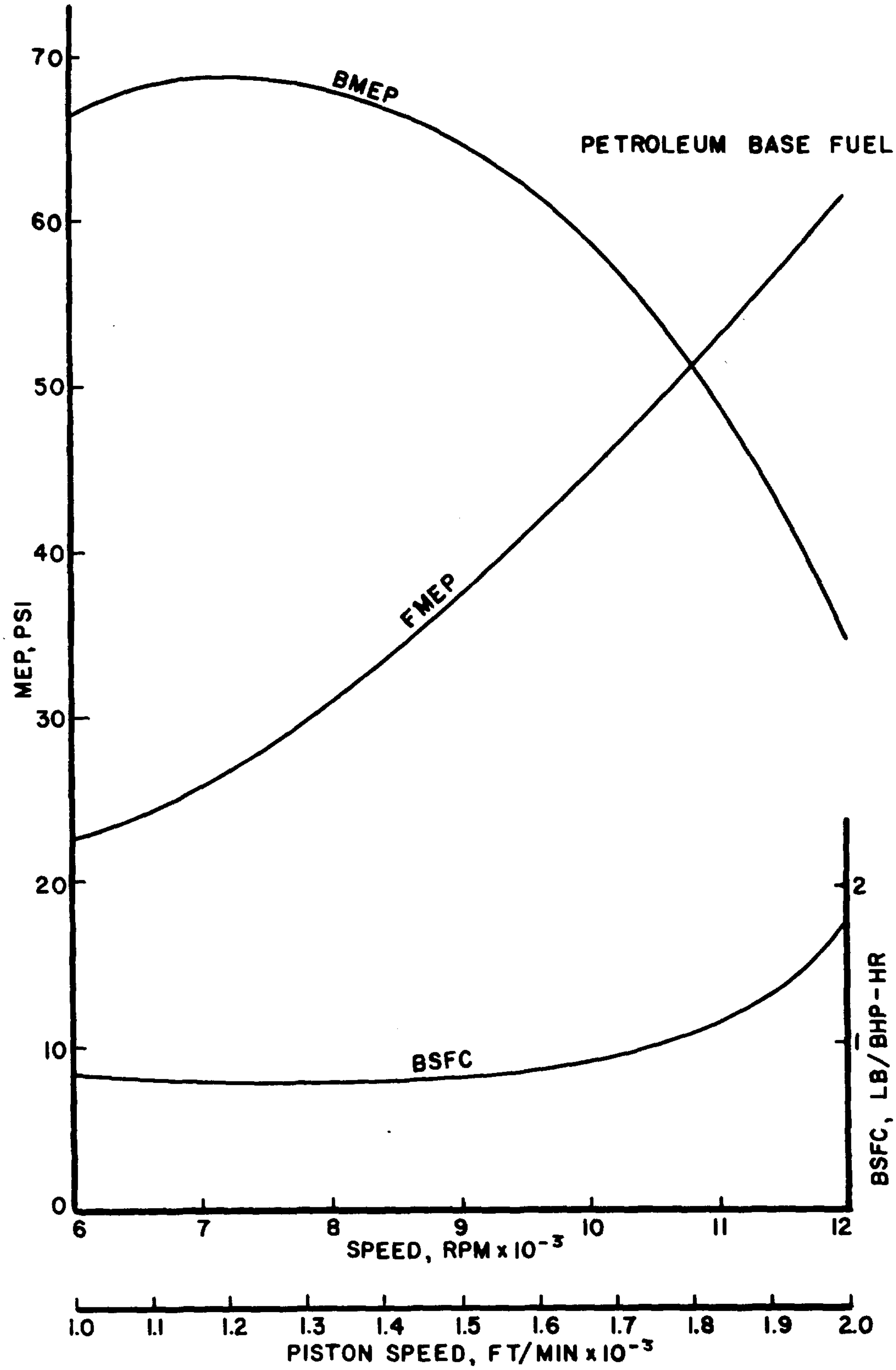


Figure No. 31. Envelope Curves for Performance of L-Head Four-Cycle Test Engine.

The superior performance of the overhead-valve engine is due in part to the fact that considerably more test work was conducted with this engine, developing it to a higher degree than the L-head engine. In addition the overhead-valve engine showed considerably lower FMEP than the L-head, particularly at the higher piston speeds. The twin camshaft arrangement of the L-head engine (Figure 22) undoubtedly contributed to its high FMEP. Since these test engines were not subjected to a development program of the type necessary to produce the maximum performance and economy, the results probably do not represent the ultimate performance possible. It is difficult to estimate what the ultimate performance for engines of this type might be, but it is certain that some improvement could be made. Improvements possible in the near future could conceivably give as much as 20 per cent improvement in BMEP, 5 per cent reduction in FMEP and 30 per cent reduction in BSFC. In practice, reproduction of the values given in the envelope curves will require some development work, and improvement on these values will certainly require additional development work commensurate with the amount of improvement sought.

After selecting an engine displacement and speed, it is possible to use the corresponding values of BMEP, FMEP, and BSFC from the envelope curves to compute complete performance specifications possible, based on the test engine experience, taking into account, of course, differences in piston speed and RPM of engines of different sizes.

2.1.3 Test Observations

The results of project test work with miniature engines have served to indicate certain trends or tendencies of interest.

2.1.3.1 Ignition Two types of ignition systems were used with the two-cycle engines, spark and glow ignition. The glow-ignition system was found to be unsatisfactory with the four-cycle engines and with two-cycle engines at low speeds (below 6000 rpm). Maximum performance with timed spark ignition was found to be from 10 to 50 per cent better than that obtained with glow ignition; however, the relative simplicity of glow ignition will make it desirable for many applications. For different engines, operating conditions, and fuels, different values of spark ignition timing were found to be optimum, varying from 32 degrees BTDC for the O. H. V. four-cycle engine at 8000 rpm to 53 degrees BTDC for the two-cycle engines at 16,500 rpm. In practice the optimum timing for a given engine, operating speed, and fuel will be determined by development tests. Starting characteristics of spark ignition engines in general are better than those of engines using glow ignition.

Compression ignition has proved to be quite unsatisfactory for most miniature engines, as is explained in further detail in Section III.

2.1.3.2 Cylinder Temperature The effect of increasing the operating cylinder head temperature of the test engines was found to reduce the maximum power slightly with no perceptible change in economy. Progressively increasing the cylinder head temperature from 320° F. to 460° F. produced a linear reduction of maximum power, the maximum power at 460° F. being 5 per cent less than at 320° F. As might be expected, higher cylinder head temperatures tend to make adequate piston cooling a difficult problem.

2.1.3.3 Crankcase Compression It is often quite difficult to achieve high crankcase compression ratios with the crankcase-scavenged two-cycle engines, because of the necessity of providing adequate clearance for the rotating and reciprocating parts. Tests with various crankcase volumes for the test engines indicated that the optimum values of the crankcase compression ratio were between 1.25:1 and 1.3:1. Crankcase compression ratios appreciably greater or less than these values produced a marked decrease in performance. The sealing of the crankcase at the rotary valve and main bearings is extremely important at the higher compression ratios, since leakage amounts to a loss of both the mixture and the additional pumping power expended. Higher crankcase compression ratios and additional pumping capacity, as might be obtained with differential size pistons, could be advantageous where asymmetrical cylinder timing is used so that short-circuiting of the mixture could be minimized and benefit gained from the supercharging effect.

2.1.3.4 Unusual Engine Configurations and Types The experience of this investigation and of others has indicated that many of the unusual engine configurations and types, appearing to offer advantages over conventional designs, do not in practice prove feasible, in their present states of development, in the miniature sizes. In particular the various methods for obtaining uniflow scavenging of the two-cycle engine would appear, on the surface, to offer promise of considerably improved economy. Results of experimentation with the poppet-valve two-cycle engine (Figure 1-E) have shown that it is inherently capable of only mediocre performance, since the valve gear imposes severe speed limitations (of the order of 6000 rpm maximum) on this engine. In this speed range and even at somewhat higher speeds, the four-cycle engine of conventional design offers considerably more promise, particularly where fuel economy is important.

An attempt by one concern to develop an opposed-piston uniflow-scavenged engine (Ref. 28) was unsuccessful because of high friction losses encountered in the linkage connecting the two pistons. This problem, it is believed, will be inherent in the opposed piston engine, and again the four-cycle engine offers a better solution at present to the problem of reducing fuel consumption.

The compression-ignition engine is an example where the experience of this laboratory would indicate that possible advantages would not compensate for the disadvantages associated with its use. A discussion of the factors involved is presented in Section III, 1.3.

2.2 ENDURANCE AND STARTING CHARACTERISTICS

2.2.1 Endurance

The data accumulated to date have been insufficient to establish statistically the endurance life potential of miniature engines. Indications are, however, that almost any practical length of operating time can be obtained with careful selection of engine type, cycle, mean piston speed, bearing pressures, etc., together with adequate design and development.

Project test results indicate that medium-to-high output two-cycle model airplane engines are capable of continuous operating life up to 200 hours, while operating on alcohol-base fuels and castor oil lubricant, with mean piston speeds up to 1500 fpm and BMEP's up to 40 psi. Another two-cycle engine, designed specifically for long life and operating at a BMEP of 27 psi and a mean piston speed of 380 fpm (at 2300 rpm), was operated continuously for 500 hours on a fuel mixture consisting of 100 parts white gasoline and 1 part synthetic lubricant.

Except when operating at very high specific outputs and/or very high mean piston speeds, engine stoppage most generally results from a failure of some component in the ignition or fuel systems; the failure of engine components, bearings, lubrication, etc. was relatively rare by comparison.

Engine component failures, particularly of connecting rods and connecting rod bearings, occurred quite frequently in the test engines when mean piston speeds above 1800 fpm were used together with BMEP's above 45 and 80 psi for the two- and four-cycle engines, respectively. Undoubtedly, operation of relatively short duration in this region and even higher would be feasible with adequate component development.

It is impossible to make recommendations as to the best piston speeds and BMEP's for the many possible applications, since the requirements of a particular application and the amount of development time available are highly important criteria in the selection of the initial design values. Final selection must be made at the designer's discretion, based on a comparison of the information presented above with current practice in larger sizes, with subsequent modification as dictated by the development test work with the engine.

2.2.2 Starting

The enigma of temperamental starting has long been associated with miniature engines, particularly of the simple two-cycle type. The experience obtained in the course of this study has not served to dispel, explain, or solve the problems. Some commercial and some test engines showed excellent starting characteristics, yet others were extremely difficult to start while showing no obvious clues to the reason in their continuous operating characteristics. It was verified in the course of the test work that starting of the two-cycle engines was, on an average, somewhat more difficult than starting four-cycle engines. Several factors could contribute to this tendency. The scavenging system of the simple two-cycle is inherently less positive than that of the four-cycle, and considerably more reliance is placed on fluid flow characteristics for completing the process. For this reason the scavenging is much less efficient at the cranking speed than at the usually much higher operating speed, and consequently the adjustment of the mixtures to within combustible limits is more difficult. Further, the presence of the lubricating oil in the fuel mixture adds to the problems of attaining satisfactory combustible mixtures and also contributing to the possibility of ignition fouling. It was found that two-cycle engines incorporating spark rather than glow ignition, and a reed crankcase valve rather than a timed rotary valve, displayed relatively superior starting characteristics.

Starting ease depends to a large extent upon the cranking speed attainable by the starting system. The systems allowing relatively high cranking speeds are of course better than hand pull or cranking systems.

2.3 GENERAL RECOMMENDATIONS

Design compromises, based on a consideration of all operating conditions and requirements imposed by the intended application, obviate the possibility of making complete and generalized recommendations as to the best type of engines for all applications. Considering certain areas of applications, however, suggestions based on project experience can be presented as an aid to the ultimate choice.

Classifying the possible applications into two groups, short and long periods of continuous operation, makes possible the following recommendations. For applications requiring only short periods of continuous operation, weight of equipment is more significant than weight of fuel consumed, and it appears that the superior prime mover would be a high-speed, high specific-power, two-cycle engine. For this application, glow ignition would provide a good, relatively simple ignition system, whereas spark ignition would give slightly superior performance and some additional complications. The free-breathing, cross-scavenged design would be best for the short operating periods. A two-cycle design using some form of loop scavenging, affording slightly better fuel economy, would be the best design where several hours of continuous operation would be required.

For longer periods of continuous operation, in excess of 25 to 50 hours, the weight of fuel and lubricant used becomes relatively more important with respect to total equipment weight. The reason for this is obvious if the total weight (equipment and fuel) is plotted as a function of operating time for various specific weights of equipment (lb/bhp) and for various values of specific fuel consumption. Figure 18 of WADC TR 53-180 (Ref. 27) is an example of a plot of this type. A moderate speed, moderate specific power, four-cycle engine, because of its better economy, appears to be the best choice of prime mover for these longer operating times. A comparison of available miniature and small two- and four-cycle engines indicates that the specific weight of the four-cycle engine, in pounds per brake horsepower, will be approximately 1.2 to 2.0 times that of a two-cycle engine. The improved fuel economy gained with the four-cycle design will provide a fuel saving over a long operating period which will more than compensate for the increased equipment weight. Spark ignition is the only satisfactory ignition system for use with the four-cycle engines.

For intermittent operation, the choice of the best type of engine design would be influenced heavily by the cycle of operation. The total length of operating time at rated load, length of time between successive start-ups, length of time at idling speeds, etc. would need to be considered. No final conclusions regarding the best engine design for intermittent operating can be made, but if the total length of operating time is considered, the same conclusions stated above for continuous operation are applicable. In addition, the four-cycle engine has demonstrated a superior performance at part load and at idling, as compared with most two-cycle designs.

3. MINIATURE ENGINE DESIGN

The detail design of miniature engines utilizes techniques and theories identical with those used for larger engine design, and, in fact, is adapted directly from large engine experience. Internal combustion engine design is an art which relies much on past development experience, in the form of empirical equations and parameter ratios, to evolve a new design of different size and/or performance characteristics. Extrapolation of design characteristics beyond the range of accepted current practice is often necessary, since most new techniques are developed on a particular size and type of engine and are then adapted to other sizes and types.

Much of the design work for a miniature engine involves such adaptation, with particular cognizance of certain characteristics which change markedly or become increasingly influential as the engine size approaches the miniature range. This discussion of miniature engine design, intended as an aid to the "prototype detail design" as discussed in Section II, 1.2.4, will be concerned primarily with emphasizing these critical adaptations of engine parameters and characteristics. By inference, it can be assumed that design subjects not treated in the discussion require only routine adaptation from existing large engine practice, according to the best present information.

Certain characteristics unique to engines in the miniature size range are, in general terms, as follows:

A. Large combustion chamber surface to volume ratio.

B. Higher rotating speeds and thus higher cycle frequencies at mean piston speeds comparable to those used with larger engines.

C. Closer absolute manufacturing tolerances and clearances.

D. Practical problems associated with metering the small quantities of fuel and lubricant required. Complete infeasibility of using direct injection of liquid fuel into cylinders.

3.1 CYLINDER DESIGN

3.1.1 Two-Cycle Scavenging

The miniature piston-ported two-cycle engine is potentially capable of only low to moderate thermal efficiencies, partly because it is at present impossible to build suitable cylinder injection pumps handling the small quantities of fuel involved, and so the fuel must be carbureted into the scavenging air. This, of course, places special emphasis on efficient porting and scavenging design. Present techniques for cross and loop scavenging are well defined in available reference material such as Schweitzer (Ref. 3), Percival (Ref. 4), and others. As for the crankcase-compression engine, information yet unavailable concerning the dynamic variation of crankcase, transfer passage, and cylinder pressures would help considerably in the application of the known techniques.

3.1.2 Combustion Chamber Design and Compression Ratios

As previously noted, the miniature engine has a considerably higher combustion chamber surface to volume ratio than do engines of larger size. This allows for higher heat transfer rates and thus the production of greater specific powers than are possible with larger engines. Since, at constant piston speed, the rotating speed increases as the size decreases, the frequency of the combustion process is also increased. However, it has been found that the extreme proximity of piston and cylinder head at top center leads to combustion problems stemming from the "quenching" action encountered; hence, for a given engine size and combustion chamber shape there is an optimum compression ratio which gives maximum power and economy. To obtain even the modest compression ratios (of the order of 6:1 to 9:1) necessary for good performance, certain difficult design problems must be solved. (Unless otherwise noted, all two- and four-cycle compression ratios quoted throughout this report are based on full-stroke displacement). The absolute clearance between the piston and the cylinder head, particularly with large bore-stroke ratios, is so small that difficulty is encountered in preventing contact. In designs involving a complicated combustion chamber shape, as in the cross-scavenged two-cycle engine, the factors which establish the optimum compression ratio are more complex and the compression ratio at optimum may be much higher than 9:1. A more complete discussion of the data which establish the existence of an optimum compression ratio are presented in Section III.

3.1.3 Cooling Design

Considerable additional investigation is needed concerning the detail design of miniature engine cooling systems. Indications are that the problems are comparable to those with larger engines. In general, the increase in cooling surface per cubic inch displacement is offset by the increased frequency of combustion cycles (at comparable mean piston speeds). As discussed in Section II, 2.1.1 at comparable outputs (i.e. identical mean piston speed and BMEP) the heat dissipation per square inch of cooling surface will be approximately the same. However the optimum cooling configuration, i.e. cooling-fin size, thickness, and spacing, will not scale down as the size is reduced, but remain essentially the same. The final design will be controlled by the space limitation on cooling fins consistent with over-all size requirements. When reduced sizes do not permit optimum cooling fins, it will be necessary to provide additional cooling blower capacity, requiring expenditure of more power for cooling purposes.

Higher specific power outputs are generally possible when use is made of cylinder and cylinder head construction materials with good heat transfer properties, such as aluminum. A cylinder liner is necessary for good-wear characteristics when aluminum is used, the liner being in the form of a steel insert shrunk in place or else porous chrome plating directly on the aluminum cylinder. When a liner is used, care should be taken to provide for the best possible heat transfer between the liner and cylinder. A fabricated steel cylinder is often preferable when complicated loop-scavenged two-cycle transfer passages are necessary.

Some manufacturers have experienced difficulty obtaining a satisfactory seal between the cylinder head and cylinder liner. Where frequent disassembly is not necessary, satisfactory results have been achieved by seating the head, without a gasket, on a narrow ridge on the cylinder liner and elastically deforming the aluminum head.

3.1.4 Bore-Stroke Ratio

It is impossible at this time to evaluate the influence of the design bore-stroke ratio on miniature engine performance. Even with large engines, generalized conclusions are subject to a great deal of controversy. It is logical to expect that the best bore-stroke ratio for a given application will be primarily a function of engine speed, other factors being equal. There is also reason to believe that the optimum bore-stroke ratio for best specific power production will not necessarily be optimum for best economy. The trend in miniature engine design is definitely to the square and over-square engines of bore-stroke ratios from 1 to 1.35. In general, the higher the engine speed and specific power output, the more justification there is for the higher bore-stroke ratios. The higher specific power outputs tend to "saturate" the larger specific heat transfer surfaces associated with large bore-stroke ratios, thus tending to retain indicated thermal efficiencies while improving mechanical efficiencies.

Until a great deal more performance results are available, bore-stroke ratios for miniature engines will have to be chosen by comparison with existing engines of similar size, speed, and application, with variation at the designer's discretion. Subsequent development to ascertain an optimum bore-stroke ratio would be very costly and time-consuming, and thus in most cases not justified.

3.2 PISTON DESIGN

3.2.1 Two-Cycle Cross-Scavenging

The design of the piston head for cross-scavenging involves a deflector to force the scavenging mixture towards the top of the cylinder and reduce short circuiting. For extremely-high-output engines an efficient aerodynamic shape is called for, which in some cases will complicate fabrication. When thermal efficiency is paramount, it is felt that a thin deflector, which will run hot, will aid in vaporization of the fuel mixture. In large engines, "directed" ports have been used to some extent to obviate the necessity for the piston baffle. In the miniature sizes, however, machining of efficient directed ports is very difficult, and they are thought to be impractical in the absence of opposing or loop-scavenging ports.

3.2.2 Piston Sealing

Piston rings of conventional design are quite common in miniature engines. Construction normally includes two compression rings located above the piston pin. Another common method of sealing is the "lapped piston" which requires no rings but relies on a very good fit with the cylinder. In the latter case, the material selected, usually cast iron, must be such that a good seal is maintained during the engine warm-up to operating temperatures. The proponents of the lapped-piston maintain that with careful design and manufacture this type piston will maintain a better seal over longer engine life than will pistons using rings. There is some evidence to support this theory, but it is evident that many of the designs on the market are not satisfactory. The requirement for a piston with a low coefficient of thermal expansion usually results in higher reciprocating weight and therefore somewhat lower speed potential.

Piston rings involve certain problems also, particularly in the piston-ported engines. Pinning of the rings against rotation about the piston is quite common in the larger engines. In the miniature sizes, however, this is seldom done, and as a result it is necessary to limit the maximum individual port angles to 35 degrees or less.

3.2.3 Materials

Piston design from a standpoint of operating temperatures is quite critical. Some manufacturers have experienced difficulty with aluminum pistons when petroleum base fuels are used and the "internal" cooling from fuel vaporization is at a minimum. When moderate to moderately high BMEP's and/or piston speeds were used, no difficulties with aluminum pistons were encountered on project test engines. As engine development increases obtainable BMEP's and piston speeds, adequate piston and cylinder cooling will become more important, and the selection of piston materials will become more dependent on the piston-head temperatures.

3.3 CONNECTING RODS AND BEARINGS

3.3.1 Connecting Rod Design

In structural design theory, miniature engine connecting rods do not differ from those of larger engines. Forged and cast aluminum alloy rods are more prevalent than in larger engines, however, the reasons being primarily economic. Connecting rod length to crank radius (l/r) ratios between 3 and 4 are preferable, but larger values are sometimes necessary for L-head four-cycle engines.

3.3.2 Piston Pin Bearings

Plain bearings are used universally for the piston pin bearings in the piston and/or rod. When the connecting rod is aluminum, it has been found desirable to use bronze inserts with the steel piston pin for reasonable bearing life. With aluminum pistons a satisfactory alternate to bushings can be achieved by using a light to moderate press-fit between the piston and piston pin. Positive location or soft piston pin pads are necessary in all cases.

3.3.3 Connecting Rod Crank Bearings

The bearing between the crank journal and the connecting rod is one of the most critical points in miniature engine design. In the smaller sizes, the "solid" rod design is preferable to the split rod, since the small fastenings are difficult to fabricate and assemble. Plain bearings have been found to be satisfactory below speeds of 10,000 rpm, provided journal diameters are large enough and lubrication is good. Plain bearings have also been satisfactory at higher speeds with certain crankcase-scavenged two-cycle engines of only moderate output and relatively short life. This is particularly true when alcohol-base fuel and castor oil lubricant are used.

The "independent roller" bearing, as used on the Dooling 61 model engine, is quite popular in engines of all sizes and speeds. They have proven very satisfactory for applications requiring high output and long life. A one-piece outer race is the most desirable but can be used only with an overhanging crank pin or an assembled crankshaft. An alternate construction utilizes a "broken" outer race where the inside separation line between the two halves of the race is "ragged" and provides for an interlocking transition surface from one half to the other.

3.4 CRANKSHAFT AND MAIN BEARING DESIGN

Proportioning of crankshafts is in general controlled by such factors as torsional rigidity and minimum bearing diameters rather than from failure strength considerations. The overhanging crank pin construction which allows the use of solid connecting rods is popular for two-cycle engines. The assembled shaft allows the use of a good air- or oil-hardened tool steel for the crank pin when it is to act as the inner race of a roller bearing.

Good results are obtained with both plain and anti-friction main bearings, the latter being preferable at speeds above 10,000 rpm. Anti-friction bearings of precision quality are preferable, since they are used at speeds considerably above their catalog ratings.

3.5 CRANKCASE DESIGN

The design of miniature engine crankcases, in general, follows routine procedures. In most cases for engine-generator application, an integral crankcase and generator housing facilitates alignment and compactness.

The crankcase internal volume and the sealing of the miniature crankcase-scavenged two-cycle engine are important to the engine operation, as discussed in Section II, 2.1.3. Crankcase compression ratios consistent with the seal capacities and the rest of the scavenging system are desirable, and are generally of the order of 1.3:1. Designing for minimum crankcase volume can be quite difficult and often requires the use of a ported-piston transfer system or light-weight filler caps opposite the crankshaft balance weight.

Four-cycle crankcase volume is not of particular importance, but care must be taken to include adequate breathing capacity to prevent the loss of lubricant through the breather passage. The wet sump design must of course include sufficient oil storage capacity.

3.6 VALVE GEAR

3.6.1 Two-Cycle Rotary Valve Design

The critical point in the design of crankcase rotary valves is the seal between the back plate and the crankcase face. For low-speed applications a spring-controlled contact between the valve and the crankcase face is adequate but is not desirable when extremely long operating time is wanted. At high rotating speeds the drag of such a system becomes prohibitive, and it is necessary to provide a plain or antifriction thrust bearing to maintain a controlled clearance between the valve and the crankcase, usually not exceeding 0.001 inches. Care should be taken to eliminate any possibility of valve deflection by the crank pin which serves as the driving pilot.

Timing of the crankcase rotary valve is dependent upon engine speed and operating characteristics. The dwell time varies from 150 to 200 degrees, closing at 30 to 70 degrees after top center. In general, it is felt that the timed crankcase valve will yield engine performance superior to that afforded by the pressure-operated or reed valve for a constant engine speed application. When idling or variable operating speeds are required, the reed valve is superior.

3.6.2 Poppet-Valve Mechanism

The design of miniature poppet-valve mechanisms does not differ in principle from that in larger engine practice, but frequencies of valve action are much higher for the miniature engines running at piston speeds comparable to those used in the larger engines. In fact, the maximum practical valve operating frequencies impose the primary limits on the attainable engine rotating speed and mean piston speed. As a consequence, particular design attention must be given to minimizing valve accelerations and reciprocating weights of the valve actuation mechanism. It is suggested that one of the more recent cam design techniques, an example of which is described in Ref. 10, be employed.

3.7 ENGINE BALANCE

The balancing of all single-cylinder engines is only partial, in that only the "primary" reciprocating forces can be balanced by crankshaft balance weights. The secondary reciprocating forces are generally not compensated, and it is desirable to reduce these as much as possible by using the lightest possible reciprocating masses, i.e., piston, rings, pin, and connecting rod. Techniques for balancing are completely described in many Internal Combustion Engine and Kinematics Textbooks, for example Ref. 21. Best results with the project test engines have been obtained with a slight overbalance, by balancing up to 66% of the reciprocating weight.

3.8 LUBRICATION

Lubrication problems in miniature engines are quite similar to those in larger engines. In crankcase-scavenged two-cycle engines when the lubricant is carbureted with the fuel, the critical component seems to be the plain connecting rod bearings, when used with relatively high bearing pressures. When the lubricant is metered independently of the fuel in two-cycle engines, particular care must be taken to provide adequate cylinder lubrication.

A wet sump with splash distribution has given satisfactory lubrication of connecting rod and main bearings, piston-pin bearings, and piston in the overhead-valve and L-head designs of four-cycle engines.

The wet sump design is limited to applications where variable engine attitude is not required.

3.9 MINIATURE ENGINE DESIGN - PARAMETER RATIOS

A survey of the design parameter ratios employed on the various available commercial miniature engines resulted in many cases in a range of values for each ratio not unlike those employed in larger engine practice. The engines surveyed were intended for a variety of applications and are included as a means of checking the conformity of new designs with current practice.

Two- and Four-Cycle Design Parameter Ratios

Bore/stroke	0.95 to 1.35
Rod length/Crank radius (l/r)	3.33 to 4.00
Crank pin dia/bore	0.23 to 0.40
Crank pin projected area/Bore area	
Plain Brg.	0.15 to 0.42
Needle Brg.	0.08 to 0.13
Main bearing dia/bore	
Plain Brg.	0.50 to 0.80
Ball Brg.	0.35 to 0.66
Exhaust-area/bore-area	0.16 to 0.55
Intake-area/bore-area	0.05 to 0.29

Two-Cycle Parameter Ratios

Exhaust timing	Opening	60° to 76° BBC
	Closing	60° to 76° ABC
Intake timing	Opening	51° to 63° BBC
	Closing	51° to 63° ABC
Blowdown		7° to 20°
Crankcase valve dwell		180° to 205°
Crankcase valve closing		45° to 70° ATC

SECTION III

FUELS & LUBRICANTS

1. FUELS

1.1 FUEL REQUIREMENTS

The fundamental requirements for miniature engine fuels are about the same as for most other types of engines and applications. The fuel should afford easy starting, produce high power and high thermal efficiency, and have no detrimental characteristics which would shorten engine life seriously. In addition an ideal fuel should have a high heating value both per pound and per gallon of fuel. It should have good storage stability for long periods of time and be readily available at a feasible cost. It also must be safe to store, transport, and use without resorting to unreasonable safety precautions. Since each of these requirements is of general interest for all engine applications, considerable information is already available in the literature on most of these properties. However, certain performance characteristics, namely starting ease, power output, and engine thermal efficiency, are functions not only of the fuels burned but also of the engine characteristics. While the starting characteristics must be determined on a more or less relative basis, the power and efficiency attained by a given fuel can be measured quantitatively in terms of the BMEP, BHP, and BSFC. Experience with different types of fuels in miniature engines has shown that antiknock quality of the fuel is almost a negligible factor. Good performance can be obtained from the low-octane natural gasoline, kerosene, and jet fuels as well as from the high-octane aviation gasoline and liquefied petroleum gases. The more volatile of these fuels seem preferable for many applications because of the greater starting ease and the cleaner running engines which result. Volatility of the fuel is an important factor in performance in that higher operating temperatures and/or improved carburetion systems are necessary to evaporate and burn the less volatile fuels.

1.2 RESULTS OF FUEL TESTS

Because of the small combustion chamber volumes, the high ratio of surface to volume in the combustion chamber, and the inherently high speeds of miniature engines, it was necessary to conduct a number of exploratory tests to establish the range of fuel properties which would give satisfactory performance. The results of several test programs conducted with various types of engines are presented here. Table 3 lists a few properties of the pure hydrocarbons used as experimental fuels in an attempt to establish the permissible ranges of octane rating and volatility for miniature-engine fuels. Hydrocarbons having a wide range of boiling points were tested in the engines to establish the effects

TABLE 3

SOME PROPERTIES OF HYDROCARBON FUELS

Fuel	Research Octane No.	Motor Octane No.	Boiling point at atm. press.
Benzene	-------	+2.75cc TEL*	176 F
Propane	+1.9cc TEL	97.1	-44 F
Methanol	+0.6cc TEL	-------	149 F
Diisobutylene	+1.7cc TEL	93.3	106 F
Isobutane	+0.17cc TEL	97.6	11 F
Isooctane	100.	100.	210 F
Butane	93.6	90.1	31 F
80-20 Blend (Isooctane, n-heptane)	80	80	210 F
n-pentane	61.7	61.9	97 F
60-40 Blend	60	60	210 F
40-60 Blend	40	40	209 F
20-80 Blend	20	20	209 F
n-heptane	0	0	209 F

*TEL - Tetraethyl lead, cc per gallon in isooctane of fuel volatility on starting characteristics and on specific fuel consumption. In addition to the fuels listed in Table 3, kerosene, 80 octane aviation gasoline, and several commercial model-engine fuels were used. A few tests were made to determine the effects on performance of various additives which are often mixed with model-engine fuels.

Figures 32 through 39 present the results of comparative fuel tests conducted with the pure hydrocarbons listed in Table 3. These tests were made with the overhead-valve four-cycle test engine operating with spark ignition and having a piston diameter of 1.25 in. and a displacement of 1.505 cubic inch. The fuels used were various blends of isooctane and n-heptane, ranging from 0 to 100 octane numbers in increments of 20, and a commercial grade of propane. While no analysis of the propane was made, commercial grades sold in this part of the country ordinarily contain approximately 70 per cent propane, 20 to 25 per cent propene, and a small amount of butane and isobutane. The octane rating of the commercial propane is usually greater than 100. Hence, these fuels have a range of antiknock ratings from 0 to greater than 100 octane numbers. The boiling points of the isooctane and n-heptane blends are between 209 and 210 F.; the propane boiling point is less than 0 F., so that the fuel is admitted to the engine in the gaseous state. This engine was lubricated with SAE 20 oil in the crankcase, and no oil was mixed in the fuel.

Figures 32, 33, and 34 show the envelope curves which indicate the maximum and minimum curves of BHP, BMEP, and BSFC for the several fuels, tested when the engine was running with compression ratios of 6:1, 8:1, and 10:1, respectively.

Some fuels produced their peak power at different engine speeds than others did, and the minimum points of the BSFC curves occurred at different speeds for the various fuels. To permit comparison, the fuels are listed in the order of the maximum power produced and the minimum BSFC at 5000 rpm. Thus, in Figure 32, the highest power was produced by 0 octane fuel and the lowest by 60 octane fuel; 80 octane gave the lowest BSFC and 40 octane the highest. Each of these fuels produced maximum power in this test at about 5000 rpm. The performance curves for the different fuels were not exactly parallel to each other, so that a listing of the fuels in the order of the power produced (or best BSFC) at some other rpm might appear in a slightly different order. However, a comparison of the power and BSFC shows no consistent trend on the basis of octane number for any of the three compression ratios reported. In general it appears that the low octane fuels produce slightly higher power than the higher antiknock ones; with BSFC data there seems to be no trend on the basis of octane number. Apparently these fuels were so nearly equal in BSFC that the variations caused by random experimental errors were greater than the inherent differences between fuels. Hence, it may be concluded that the octane rating of the fuel has almost no significance in determining the performance of miniature engines of this type.

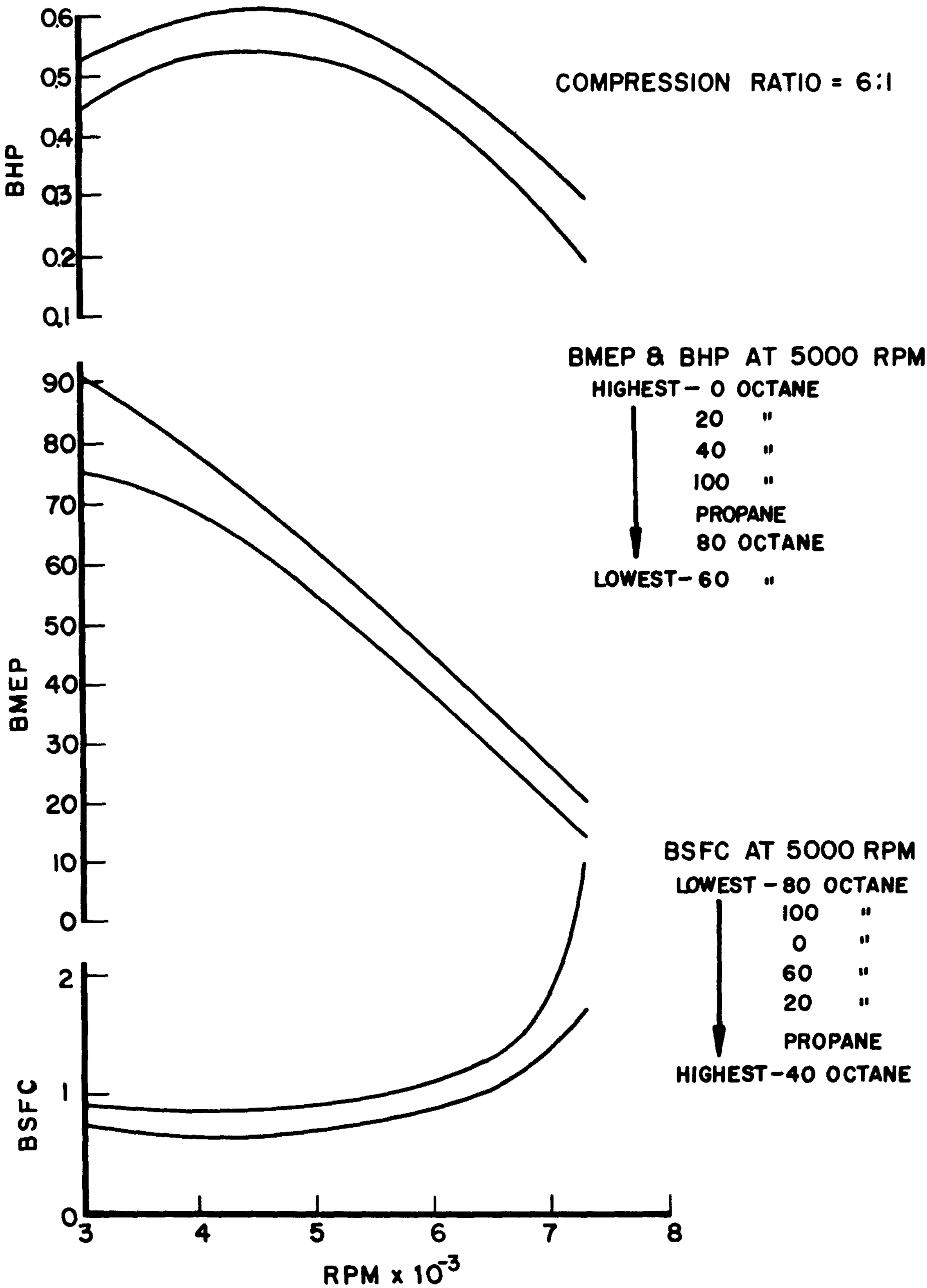


Figure No. 32. Envelope Curves for Performance of Throttled Overhead - Valve Test Engine for Various Fuels at 6:1 Compression Ratio.

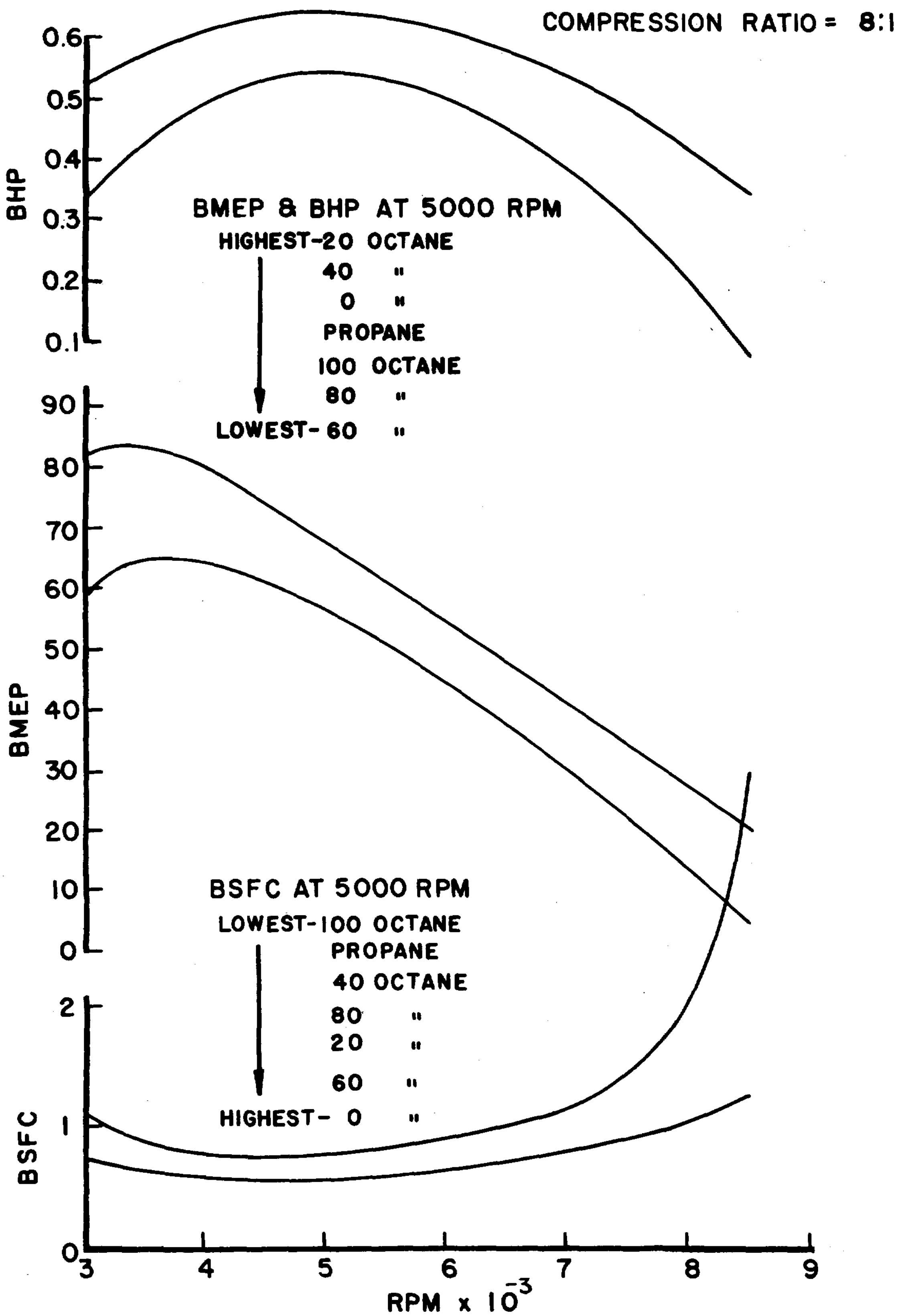


Figure No. 33. Envelope Curves for Performance of Throttled Overhead-Valve Test Engine for Various Fuels at 8:1 Compression Ratio.

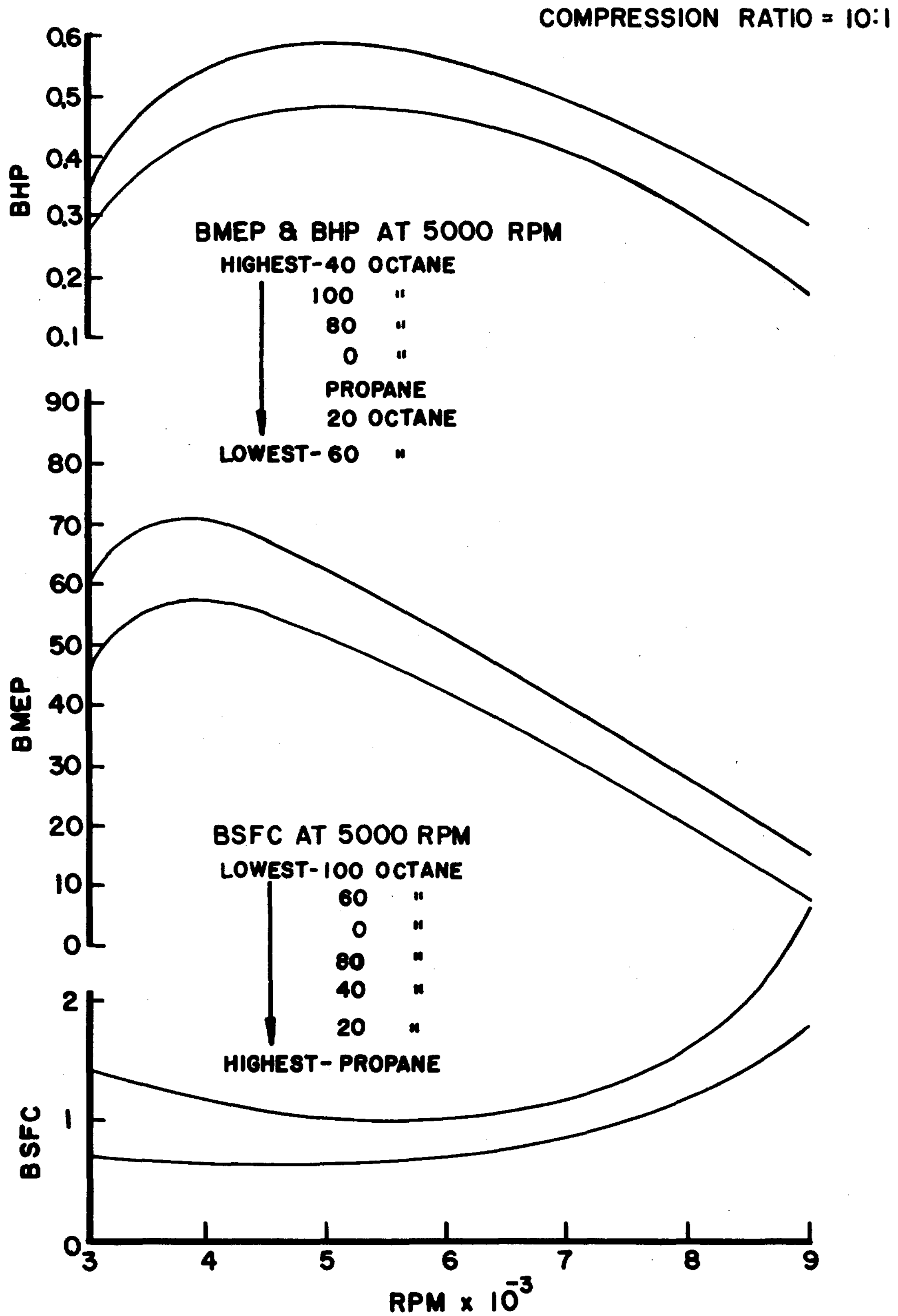


Figure No. 34. Envelope Curves for Performance of Throttled Overhead-Valve Test Engine for Various Fuels at 10:1 Compression Ratio.

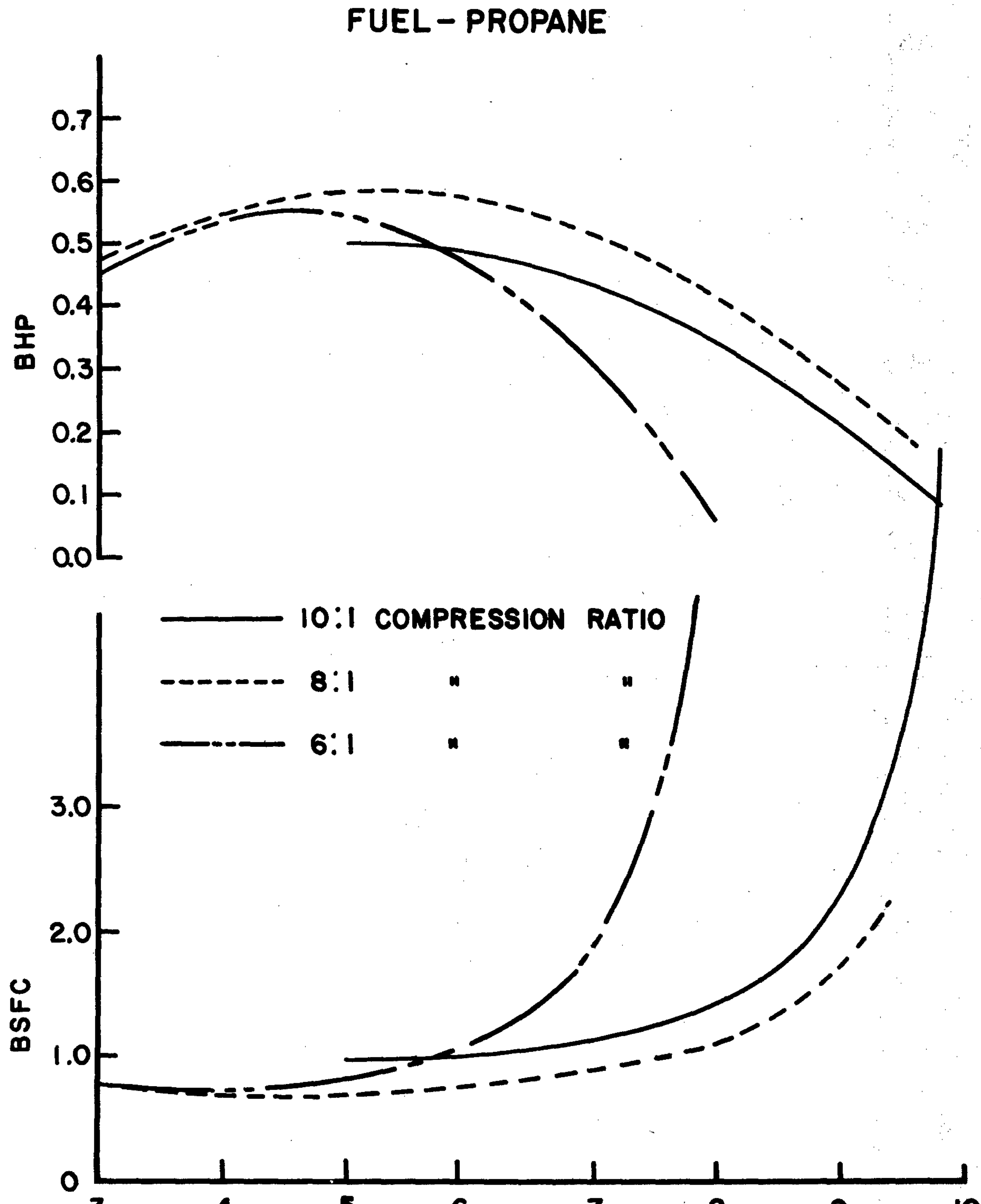


Figure No. 35. Performance Curves for Throttled Overhead-Valve Test Engine - Propane Fuel at Various Compression Ratios.

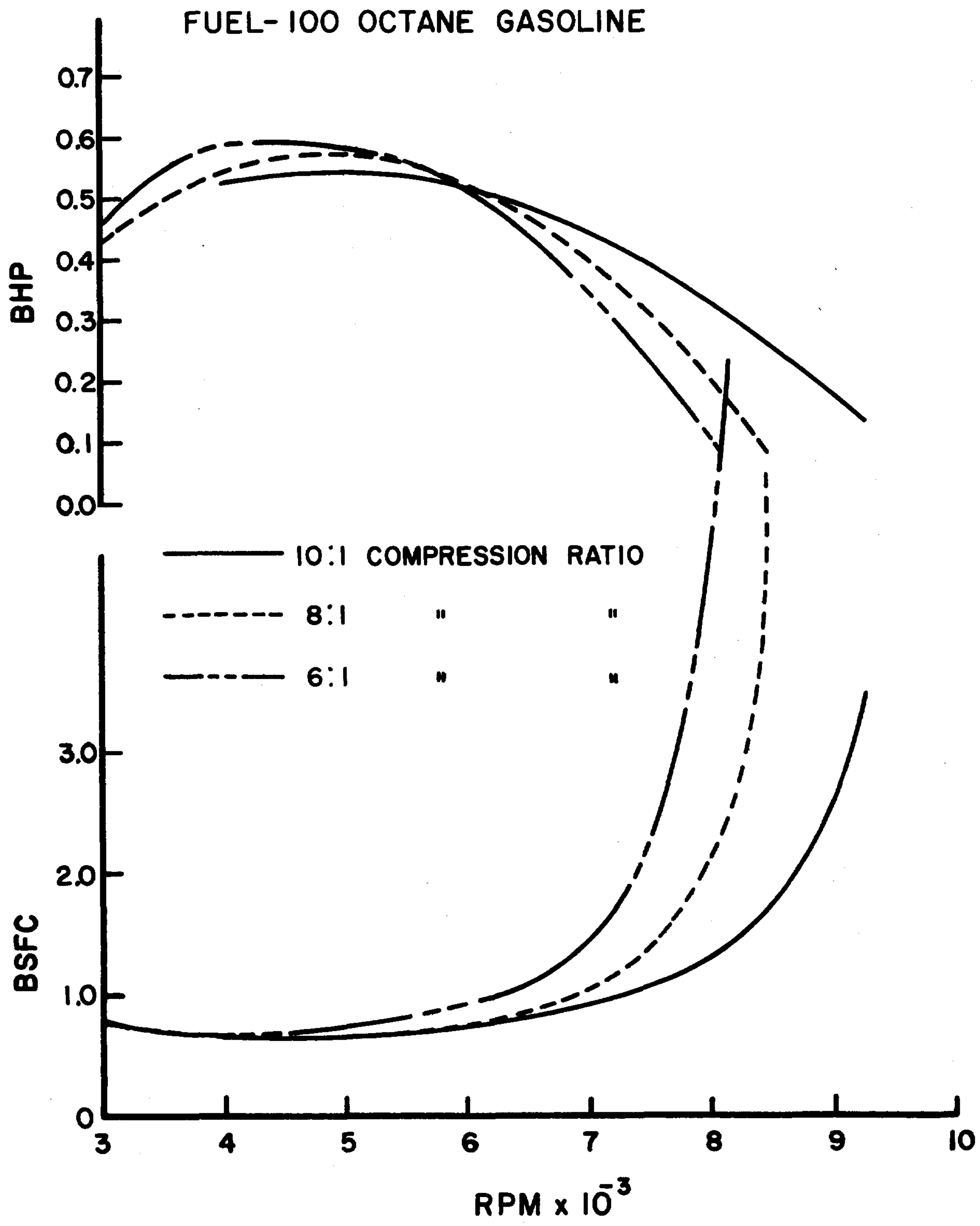


Figure No. 36. Performance Curves for Throttled Overhead-Valve Test Engine - 100 Octane Gasoline at Various Compression Ratios.

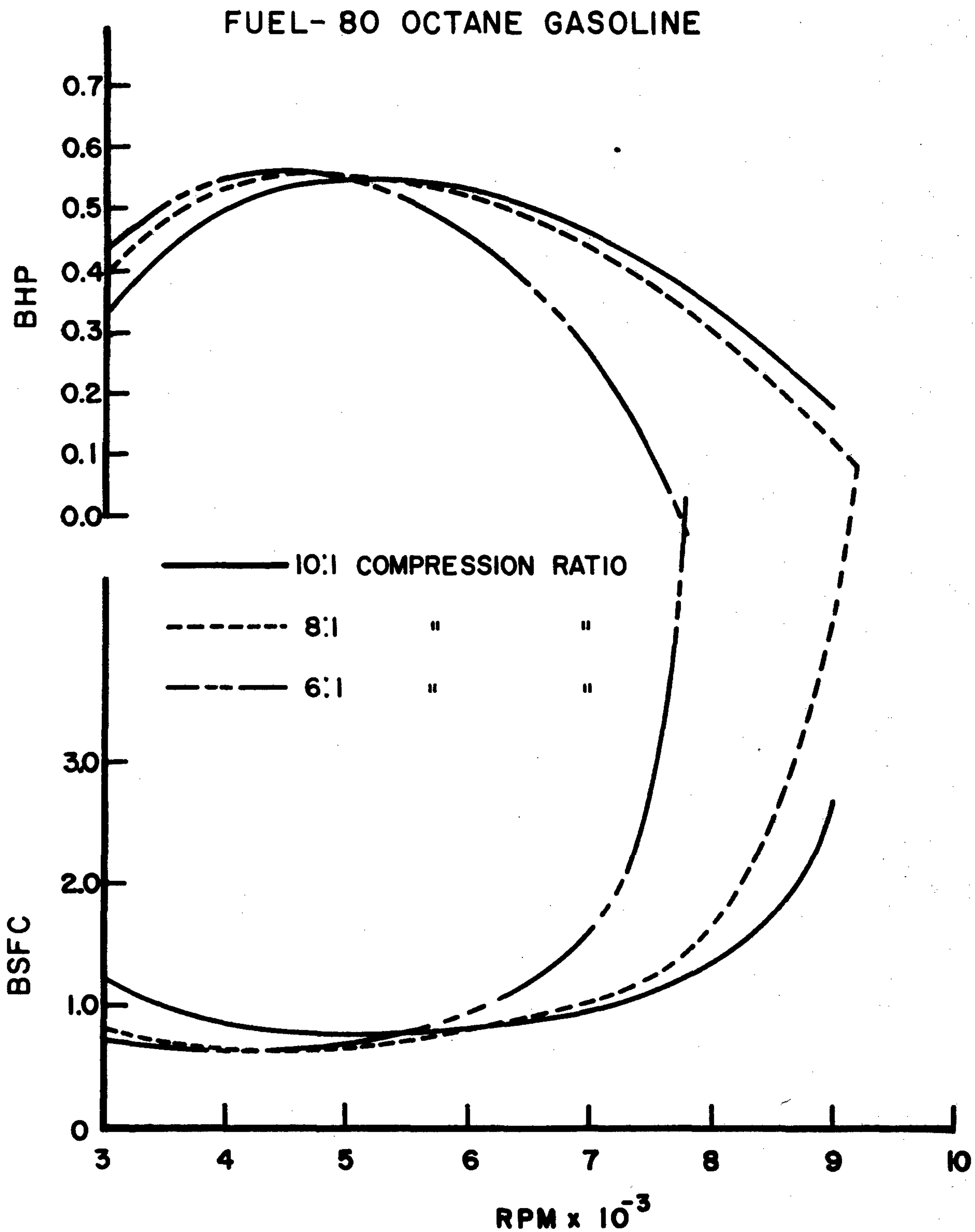


Figure No. 37. Performance Curves for Throttled Overhead-Valve Test Engine - 80 Octane Gasoline at Various Compression Ratios.

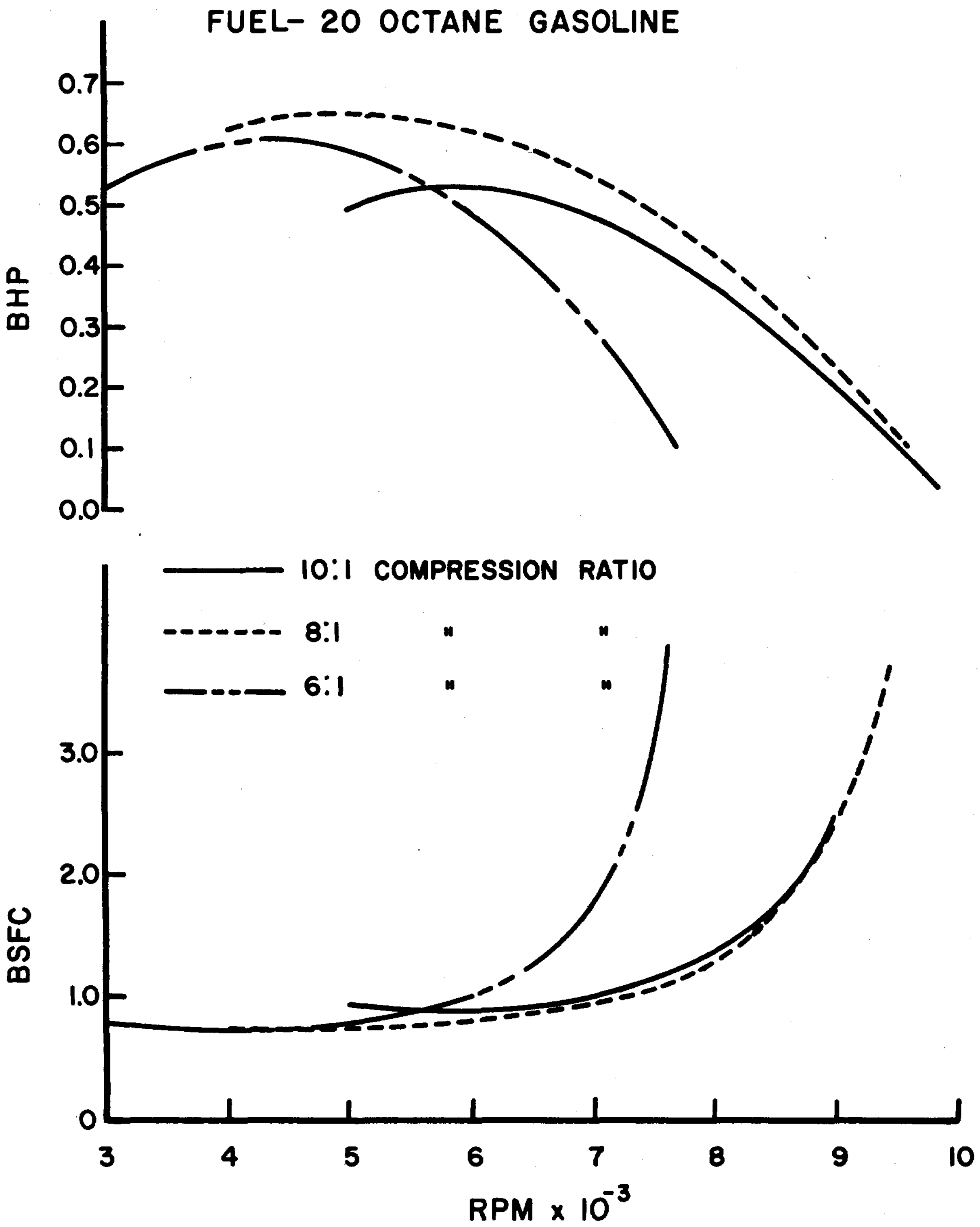


Figure No. 38. Performance Curves for Throttled Overhead-Valve Test Engine - 20 Octane Gasoline at Various Compression Ratios.

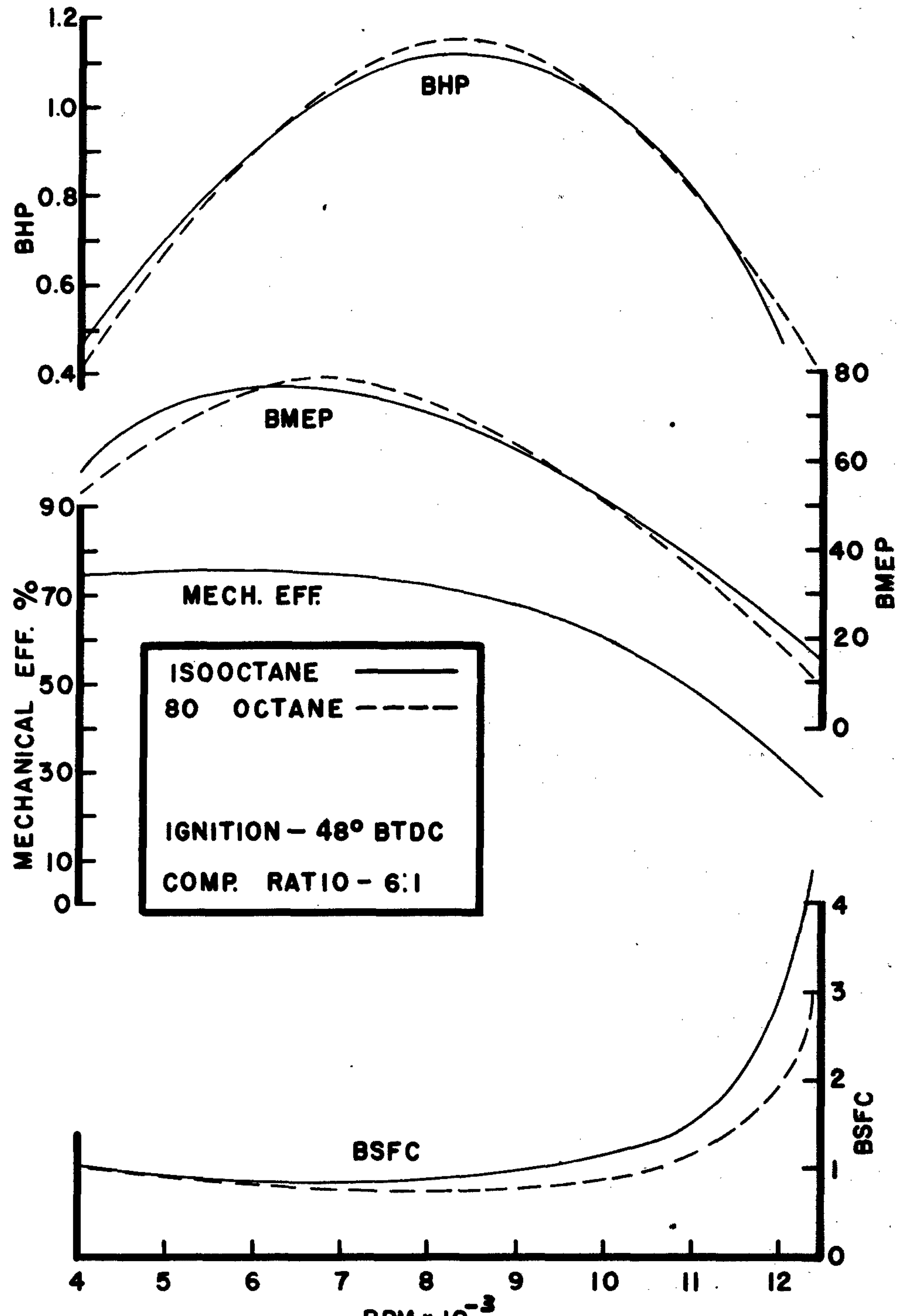


Figure No. 39. Performance Curves for Unthrottled Overhead-Valve Test Engine with 100 Octane and 80 Octane Gasoline Fuel.

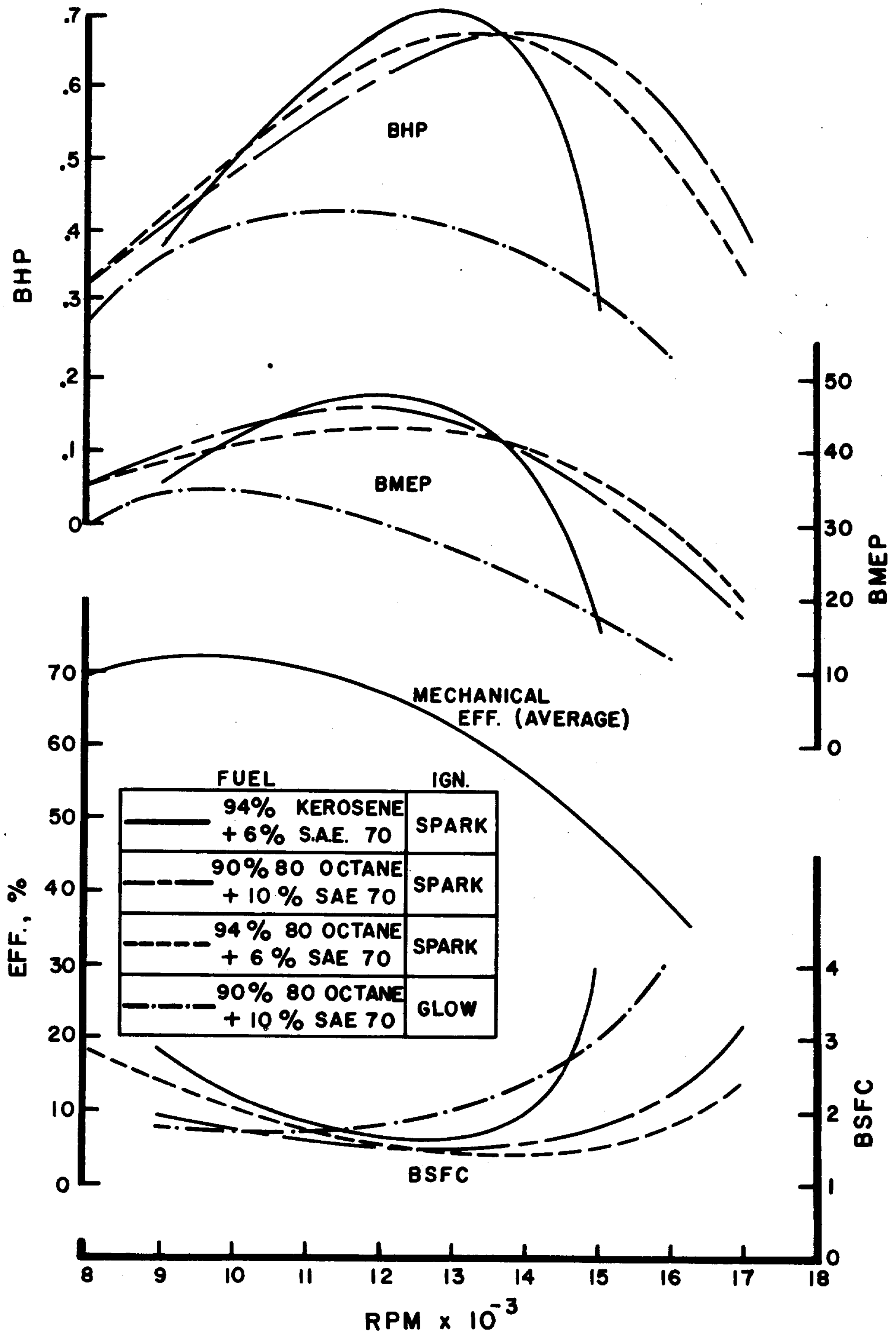


Figure No. 40. Performance Curves for Loop-Scavenged Test Engine with Various Fuels and Ignition Systems.

A comparison of Figures 32, 33, and 34 also shows very little variation in the BHP, BMEP, or BSFC with compression ratio. Figures 35 through 38 indicate the variation of engine performance with change in compression ratio for propane, isooctane, 80 octane blend, and 20 octane blend fuel. The curves show that performance is slightly better at a compression ratio of 8:1, and that the 10:1 compression ratio is no better than the 6:1 at low speeds and only slightly better at higher speeds. Thus, for this particular miniature engine there is an optimum compression ratio and even high antiknock fuels do not perform any better in it at compression ratios greater than the optimum.

For the tests just described, the four-cycle engine was throttled to about 60 per cent of its maximum power by throttling the fuel-air mixture with an orifice placed just ahead of the inlet valve. It was found that at maximum power this particular engine would develop frequent mechanical failures in the component parts. To preserve the engine, most of the tests were conducted at what was, in effect, a part load for the engine. A few results obtained at full load before failures occurred are reported in Figure 39. At full admission of air the engine peaked at about 9000 rpm and developed almost twice the power shown in Figures 32 through 38, but the BSFC and BMEP for the two conditions of operation are almost identical. Thus, it is possible to operate miniature four-cycle engines at half load to full load and obtain BSFC values of the order of 0.6 to 0.8 lb per bhp-hour. Again the octane rating and boiling point of the hydrocarbon used apparently did not affect the performance of the engine. A number of other tests conducted at full load with a wide variety of fuels showed the same results. Tests at both full load and part load indicated that detonation does occur in this size miniature engine (1.25-in. bore) at compression ratios of 8:1 and 10:1 with the 0, 20, and 40 octane fuels. This is most noticeable, of course, at speeds of 4000 to 5000 rpm, rather than at the higher speeds.

Studies similar to those just described were also conducted with a two-cycle, cross-scavenged engine. This engine used glow-plug ignition, 0.725-in. bore, and 0.724-in. stroke, with a displacement of 0.299 cubic inches. The fuels used were benzene, diisobutylene, isooctane, 80 octane, n-pentane, and 40, 20, and 0 octane blends of isooctane and n-heptane. Ten per cent of SAE #70 lubricating oil was added to each fuel. The results of these tests were even less consistent than the four-cycle engine tests in regard to showing a correlation between octane rating of the fuel and peak power developed. Tests at five different compression ratios indicated that random errors in testing were more significant than the octane rating of the fuel. The compression ratio was calculated on the basis of the total piston displacement; hence it is higher than the actual compression ratio starting from the point of exhaust port closing. Even then, for this particular engine the optimum compression ratio for maximum power was found to be between 6:1 and 7:1. The power produced by all fuels at 9:1 compression ratio was at least 15 per cent lower than the power at 7:1. In changing the compression ratio, the shape of the cylinder head and piston crown were altered from the original design. It was found that this change in combustion chamber shape had a greater effect on the maximum power attainable than the change in compression ratio from 5:1 to 9:1. This effect on power was true for all of the fuels tested. There was a noticeable increase in power with the low octane fuels at the lowest compression ratio, but this trend was not consistent at the higher compression ratios. There was about 10 per cent variation in power between the best and worst fuels at 7:1 compression ratio, but 0, 20 and 40 octane, n-pentane, and benzene all gave about the same maximum power. It does not appear that either fuel antiknock quality or high volatility, as obtained in n-pentane, is of extreme importance for miniature two-cycle engine fuels, except for the advantages of high volatility during starting. All of the above tests were run with a glow plug turned on. At the lowest compression ratio only the low octane fuels and benzene would run smoothly, but at the 9:1 compression ratio all of the fuels burned smoothly with the glow plug turned on. When the glow plug was turned off (electric current was turned off but the plug remained in the cylinder head) only the 0 and 20 octane, n-pentane, and benzene would ignite at 5:1 compression ratio. As the compression ratio was increased, the fuels ignited better and burned more smoothly with the glow plug off. Operation with all fuels, though, was better with the glow plug heated electrically. All of the fuels except n-pentane gave good starting characteristics with a warm engine. The highly volatile pentane apparently tended to "flood" when the engine was already warm from previous runs. No extensive tests of starting with a cold engine were made, but pentane should have an advantage over the other fuels for cold engine starts, because of its high volatility.

Figure 40 presents the results of tests conducted with gasoline and kerosene in the two-cycle, loop-scavenged test engine having a 0.890-in. bore and a displacement of 0.466 cubic inch. As in the four-cycle engine tests, there is little difference in the power and BMEP developed by gasoline and kerosene when the two-cycle engine is operated with spark ignition. The BSFC for kerosene is somewhat higher and is influenced more by engine speed than that for gasoline. There is a good possibility that this higher BSFC is a result of the lower volatility of kerosene and that a portion of the fuel does not evaporate or does not mix uniformly with the air in time for complete combustion. With glow ignition using 80 octane aviation gasoline, the power, BMEP, and BSFC are all less satisfactory than with spark ignition. The same comparison of performance with spark ignition and glow plug ignition has been verified with several other engines and fuels. It is felt that the lower BMEP and higher BSFC for this engine, as compared to the four-cycle engine tests reported in this section, are inherent in the two-cycle design, in which some fuel has to be wasted in scavenging the cylinder, and the effective expansion ratio is less than in a four-cycle design.

Tests with a two-cycle, loop-scavenged, engine operated as a diesel (compression ignition) were also made. The engine had a 1.015-in. bore, 0.607 cu. in. displacement, and utilized crankcase compression. The compression ratio was calculated to be 26:1 on the basis of the full piston displacement. The following fuels mixed with 10 per cent SAE #70 lubricating oil were tried, but the engine would not start with any of them: 80 octane aviation gasoline, n-heptane, diisobutylene, benzene, isooctane, and n-pentane. Methanol with 20 per cent castor oil and methanol with 20 per cent ether and 10 per cent castor oil also failed to start the engine. Various mixtures of kerosene and ether, with 10 per cent SAE #70 oil, would burn fairly smoothly. At least 9 per cent ether in the kerosene was needed before the mixture would burn well enough to run the engine. A mixture of 45 per cent kerosene, 45 per cent ether, and 10 per cent SAE #70 oil gave the maximum power attained, but the BSFC was 9.1 lb/bhp-hour. This same fuel mixture would operate the engine at a higher rpm with glow ignition than with straight compression ignition operation, but the fuel consumption was still exorbitant.

Figure 28 shows the lowest values of brake specific fuel consumption obtained with alcohol-base fuels. These tests included methyl alcohol and castor oil mixtures as well as various model-engine fuels. Many of the latter included different types of ignition and combustion additives, such as nitromethane, acetone, etc. While some of the model fuels produced better power and lower BSFC than others, the BSFC values with the best alcohol fuels were much higher than those obtained with the petroleum fuels.

1.3 DISCUSSION AND CONCLUSIONS

From the results presented in the preceding section, it appears that the antiknock qualities of the fuel hold little importance in either the four-cycle or in the two-cycle engines, whether glow plug or spark ignited. Thus, good performance could be obtained from low octane liquids such as natural gasoline and kerosene or high-octane fuels such as aviation gasoline and liquefied petroleum gases. The lower octane fuels usually show a very small increase in power output over the higher antiknock fuels, but the specific fuel consumption for different antiknock fuels seems to be the same.

The boiling point of the fuel does not appear to have much effect upon performance, provided there are sufficient temperature, distance, and time to evaporate the fuel and to permit the vapor to mix with the air. These qualifications of course will change with the engine type, design, and operating conditions.

The optimum BSFC shown in Figures 35-38 lies between 0.6 and 0.75 lb/bhp-hour for propane and for 100, 80, and 20 octane fuels. This would seem to indicate that fuels with a boiling point of 210 F.

achieve complete evaporation in time for complete combustion. If the BSFC of the propane gas were appreciably lower than those for the liquid fuels, it might be suspected that the liquid fuels were not being evaporated. Since 210 F. is approximately the 50 per cent evaporation point for most aviation gasoline, it might be concluded that with gasoline there is no appreciable loss in power or fuel due to poor evaporation. Specifically, these results show that hydrocarbons having boiling points up through the range encountered with aviation gasolines will evaporate satisfactorily and yield good power and BSFC in simple designs of four-cycle and two-cycle engines.

These results have been obtained in a four-cycle engine in which the fuel was admitted through a simple needle valve, at a pressure of a few inches of fuel, and mixed with air in a manifold about 1.5 inches long. The same evidence of satisfactory evaporation has been found in two-cycle engines in which the fuel was mixed with the lubricant and admitted to the engine through the crankcase. With fuels less volatile than gasoline, such as kerosene and jet engine fuels, it appears that poor evaporation may account for the increase in BSFC usually found when performance with kerosene is compared with that for gasoline. Since 0, 20, and 40 octane blends of isooctane and n-heptane produce the same power as gasoline, it does not seem logical that the reduced performance with kerosene is a result of the low antiknock hydrocarbons which the kerosene possesses so abundantly.

When operation at temperatures lower than room temperature is considered, then the lighter and more volatile fuels display a distinct advantage, especially for starting. Also, operating experience with many types of present commercial small engines has shown that fewer cylinder deposits and less exhaust port fouling exist when fuels having higher volatility are used.

Alcohol-base fuels do not seem to offer any particular advantage over petroleum fuels for miniature engine-generator applications. The alcohol fuels permit somewhat higher power per cubic inch of engine displacement, which has an advantage for model engines in racing competition, but the BSFC is several times as great as for petroleum fuels. For a period of operation of more than a few minutes, the weight of petroleum fuel used would be appreciably less than for an alcohol fuel, because the heating value of alcohol per pound is only about 2/3 that of petroleum fuels. The various petroleum fuels themselves differ little in heating value per pound.

Insufficient operating experience with different engines and commercial fuels is available to make definite conclusions regarding the effects of the various additives currently present in many brands of gasoline. However, there is no evidence that miniature engines differ appreciably from larger ones in the tendency for lead and carbon deposits to foul the cylinder, spark plug, and exhaust passages.

The fact that miniature engines have an optimum compression ratio which establishes a maximum power and efficiency apparently is related to a quenching effect of the flame in the very small clearance volumes of these engines. While larger engines continue to show an increase in efficiency and power output with an increase in compression ratio, the experience of this laboratory and certain others working in the small-engine field has indicated that optimum compression ratios do exist. It has further been shown that the compression ratio that will give maximum performance for a given engine is influenced very strongly by the shape of the combustion chamber. Some manufacturers of small commercial engines have found that the optimum compression ratio could be increased, thus increasing the power and engine economy, by making the cylinder head recess conical shaped, or modified hemispherical. Undoubtedly this design would reduce the quenching effect to a minimum. Experience of this laboratory and others to date indicates that the optimum compression ratio for many engines of 0.75- to 1.5-inch bore is probably between 6:1 and 9:1, calculated on the basis of the full piston displacement. However, the two-cycle, loop-scavenged engine referred to in connection with Figure 25 gives very good performance with a compression ratio of 16.3:1 based on full stroke, or 12:1 based on the time of port closing. It may be that several unique factors of cylinder and combustion chamber geometry permit this particular design to reach optimum performance with an unusually high compression ratio. No tests were conducted at this laboratory to determine the optimum compression ratio for this engine. That some two-cycle engines with ports for admission and exhaust have about the same optimum compression ratios as found for some four-cycle engines using valves is further evidence that the quenching effect is the dominating factor in the phenomenon. This same theory of quenching effect explains why miniature compression-ignition engines are generally difficult to start, have low specific power and poor economy, and are unusually sensitive to operating adjustments. The low compression ratios which yield optimum operation are too low to provide good compression ignition; and the small clearances necessary to obtain high compression ratios for good compression ignition quench the flame and permit the gases to cool as soon as they autoignite.

It may be predicted with some confidence that the BSFC values of 0.6 to 0.75 reported in Figures 32-39 are close to the minimum values that can be attained with miniature engines with a reasonable amount of development time expended. The value of 0.6 pound of fuel per bhp-hour is almost as low as is achieved in most automobile engines today. The large automobile engines may be improved by using higher compression ratios, but the miniature engine is handicapped by the high ratio of surface area to volume of the combustion chamber. Since the mechanical efficiency of the four-cycle engine used for these tests was about equal to that of present highly developed automobile engines, it does not seem likely that any great improvement in fuel economy is possible.

1.4 GASEOUS FUELS

In addition to the various liquid fuels just discussed, mention should be made of the merits of gaseous fuels, such as hydrogen, acetylene, methane, ethane, and the C_3 and C_4 hydrocarbons commonly known as liquefied petroleum gases or LPG. Each of these gaseous fuels should have better starting characteristics than any of the liquid fuels, because the problems of evaporation would be eliminated. Also, for miniature engines the metering problem of maintaining the desired fuel-air ratios might be less difficult because larger orifices could be used with gases than with liquids. Many of the problems concerning service life of the engine, where the fuel contributes as a limiting factor, would be less severe or even eliminated with most gaseous fuels, i.e. fouling of the exhaust ports with carbon deposits, fouling of glow plugs or spark plugs with carbon, and scouring of piston skirts and cylinder walls with unevaporated liquid fuel acting as a solvent to wash away the lubricating oil. However, with all gaseous fuels except LPG the storage problem far offsets the advantages listed above; it must be concluded that all fuels which must be stored as a gas are impractical for miniature engine-generator applications.

1.4.1 Liquefied Petroleum Gases

The data listed in Table 4 permit a ready comparison of the properties of the principal hydrocarbons which appear in LPG. Average commercial propane is composed of approximately 60 per cent propane, 30 per cent propene, and 10 per cent butane, ethane, and other hydrocarbons of C_2, C_3, and C_4 molecular weight. Thus, the average commercial propane has properties similar to propane and propene. A typical commercial butane is composed primarily of butane, butene, and isobutane, but usually has enough of the C_3 hydrocarbons to give vapor pressures slightly greater than for the C_4 hydrocarbons alone.

Preliminary tests using both commercial and pure propane in miniature engines indicate that good starting, high power output, and low specific fuel consumptions are possible with these fuels (Figure 35). Performance with LPG is at least as good as with any liquid fuel tested. Starting characteristics might be much better than with many liquid fuels.

Some aspects of the use of LPG for miniature engine fuel are discussed in the following paragraphs. Experience with larger LPG engines has shown that the service life between overhauls is about twice that of gasoline engines, primarily because the LPG does not wash the lubricating oil off the cylinder walls and piston and has clean-burning characteristics. The oil consumption is about one-half that for engines burning gasoline, and engines can be idled for hours without damage. The weight of LPG required for a given operating time would be equal to or somewhat less than that of a corresponding gasoline, kerosene, or jet engine fuel supply. The heating value per pound of LPG is nearly 10 per cent greater than for the commercial liquid fuels. Assuming the overall thermal efficiency to be equal for the liquid and gaseous fuels, the total weight of fuel required would be lower with LPG. The design

TABLE 4

PROPERTIES OF CONSTITUENTS OF LIQUEFIED PETROLEUM GAS

Fuel	Formula	Vapor Pressure, psia					Btu/lb	Btu/gal of liq.	Lb/gal liq.60F
		130F	100F	70F	0F	-40F			
Ethane	C_2H_6	>1000	>700	563	206	113	22,300	70,200	3.15
Propane	C_3H_8	275	189	125	38	16	21,700	91,000	4.24
Propene	C_3H_6	323	226	151	48	20	21,000	91,600	4.35
Butane	C_4H_{10}	81	52	31	7	2	21,300	103,800	4.87
Isobutane	C_4H_{10}	110	72	45	12	4	21,300	99,800	4.69
Butene	C_4H_8	97	63	38	9	3	20,800	104,500	5.01
Pentane	C_5H_{12}	27	16	12	2	0	21,100	111,000	5.26
Ave. comm. propane		301	207	139	43	18	21,560	91,500	4.24
Ave. comm. butane		112	74	46	13	5	21,180	102,600	4.84

of a good, flexible carburetor to use LPG might be easier to achieve than one for liquid fuels, since a larger orifice could be used than when metering liquids, and the high vapor pressure of the fuel itself would aid in good metering and mixing. The volume of the fuel tank for LPG would be about 25 per cent larger than for gasoline or kerosene, assuming the same over-all thermal efficiency. This might not be a penalty, as some reduction in fuel consumption at part load might be gained with a well-developed LPG engine.

At first thought it would appear that storing liquid propane or butane under pressure would impose a severe penalty on the system because of an unduly heavy fuel tank. Analysis of the problem shows that a 100-watt engine-generator set having an over-all thermal efficiency of 10 per cent (which may be conservatively low) would consume LPG at the rate of 3.9 lb per 24 hours, or 0.8 gallons per 24 hours, or approximately 1.6 lb/kw-hr. Allowing for the high coefficient of thermal expansion of LPG and for storage at the extreme temperature of 130 F., the weight of the fuel tank itself for 0.8 gallons of liquid butane would be 7 lb or less. Even to provide fuel for two or three days of continuous operation, a properly designed fuel tank need be only slightly heavier than would be required for structural rigidity alone.

A specification for LPG fuel could be written to provide reasonable vapor pressures for the operating conditions anticipated. If operation at extremely low temperatures may be required, the percentage of the light hydrocarbons of ethane, propane, and propene could be increased to provide sufficient vapor pressure to operate the carburetion system properly. On the other hand, if unusually high storage or operating temperatures were expected, the percentage of the heavier hydrocarbons such as butane and pentane could be increased to keep the vapor pressures within reasonable limits and thus reduce the weight of the fuel tank needed.

Another interesting feature of the potentialities of LPG for miniature power sources is the storage stability of these light hydrocarbons. The usual problems of gum formation in the fuel tank, which may plug fuel lines, carburetor orifices, and do other damage to the engine, should be nearly nonexistent with LPG. Thus, here would be a very good fuel for emergency equipment that must be stored for long periods of time with a minimum of attention but must function properly when used.

While insufficient experience with LPG in miniature engines is available to verify its over-all merits, any advanced planning for miniature power sources for specific applications should consider the very favorable potentialities of such a fuel.

1.4.2 Methane or Natural Gas

Methane, the primary constituent of natural gas, has a critical temperature of -116 F. Therefore, this fuel always would have to be stored in the vapor state and would require exorbitantly large and heavy fuel tanks. For example, a 100-watt engine-generator set having an over-all thermal efficiency of 10 per cent would need 3.4 lb of methane per 24 hours, or 81 cubic feet of gas at 14.7 psia and 60 F. Stored at 100 psig. and 60 F., the volume of this amount would be 10.4 cubic feet, and at 1000 psig. and 60 F. it would be 1.2 cubic feet. From these data it is evident that miniature engine fuel must be stored in the liquid state to keep the volume and weight of the system practical.

1.4.3 Hydrogen

Since the critical temperature of hydrogen is -400 F., it must be stored in the gaseous state and has the same limitation as methane - the fuel tank would be too large and heavy. The higher heating value of hydrogen is much higher than for hydrocarbon fuels, 60,960 Btu/lb hydrogen. Even so, for a 100-watt power output for 24 hours, with a 10 per cent over-all thermal efficiency, the fuel required would be 1.6 lb and would have a volume of 300 cubic feet at atmospheric pressure and 60 F. At 1500 psig. a fuel tank volume of 3.0 cubic feet would be needed. Even for a one-hour run the volume of fuel tank required would be 0.36 cubic feet at 500 psig. or 0.12 cubic feet (about the size of a gallon container) at 1500 psig. Hence, the use of hydrogen as a regular fuel is impractical for portable power sources. Some attempts have been made to develop a portable hydrogen generator to supply miniature engines, but no successful results have been attained to date.

Because hydrogen will burn in air in almost any mixture strength (limits of inflammability are 4.0 to 74.2 per cent hydrogen by volume in a hydrogen-air mixture), hydrogen has been used with some success as a fuel to facilitate starting. In a capsule having reasonable size and weight may be stored enough hydrogen to operate a small engine for approximately one minute. In this short time the engine temperature becomes high enough to evaporate and burn another fuel. The use of hydrogen in this capacity may continue, especially in equipment designed for automatic starting, since it definitely aids in this application.

1.4.4 Acetylene

Acetylene has a heating value of 21,460 Btu/lb, about the same as other light hydrocarbons. Hence, assuming the same thermal efficiency, the weight of acetylene fuel burned per day in an engine would be about the same as for other hydrocarbon fuels. For a 100-watt power unit with

10 per cent thermal efficiency, the weight of acetylene required would be 3.8 lb per 24 hours, or 56 cu ft of gas at one atmosphere and 60 F. Like the limits of inflammability for hydrogen, those of acetylene are very wide, 2.5 to 80.0 per cent of acetylene by volume in a fuel-air mixture. Hence, for starting purposes it would be very useful, since the fuel-air mixture could be in almost any proportion and still ignite. However, acetylene has the same storage limitations as methane and hydrogen, along with a number of peculiar characteristics that make it an undesirable fuel for miniature engines.

Acetylene is one of the few hydrocarbon fuels that liberate heat upon decomposition. Whenever it is subjected to excessive heat or pressure, the bonds uniting the carbon and hydrogen atoms break down. Above 1435 F. or above 30 psig. the gas is unstable, and additional heat may start an explosive decomposition of the entire quantity of gas. This decomposition could occur in storage or in the cylinder of an engine prior to completion of the combustion process. As a result there is a strong tendency for acetylene to form heavy carbon deposits in the cylinder and exhaust ports by decomposition without subsequent complete burning. These carbon deposits would limit the service life of any miniature engine seriously. The accepted safe practice in the welding field prohibits the storage or use of acetylene above 15 psig. Thus, the volume of fuel in storage for engine use would always be prohibitively large. For welding use, the storage volume is reduced by storing the acetylene in a tank containing a porous filler and acetone, which absorbs about 25 times its own volume of acetylene. Even this technique would be unsuitable for miniature engine use, as evidenced by the fact that the standard containers of acetylene for welding hold only 40 to 300 cu ft of gas at atmospheric pressure. A light-weight, portable, acetylene gas generator appears unfeasible, as a relatively large amount of water is needed to keep the operation of the acetylene generator stable. In present generators producing acetylene for welding, about 2 cu ft of acetylene is produced per hour per pound of calcium carbide charged. Acetylene in contact with any alloy containing more than 67 per cent copper may form acetylides, which are violently explosive and can be detonated by a slight shock.

2. LUBRICANTS

Operating experience on this project, and the experience of many other investigations with different types of small engines, indicates that there is no simple or general conclusion to be reached regarding the best lubricant for miniature engines. The type of engine design, operating conditions, fuel used, and manufacturing tolerances of each particular engine all influence the serviceability of any lubricant used. As reported by the Cooperative Research Council in CRC L-32-653, extensive studies of spark-plug fouling in outboard motors showed that the difference between motors of different makes run with two widely different types of test oils was greater than the average difference (where all motors were considered) between the two oils in any given motor. Of these particular special oils for outboard motors, the oil which showed the lesser tendency for spark plug fouling gave greater varnish deposits on the piston skirt and caused more ring sticking; combustion chamber and exhaust port deposits were essentially the same for the two oils.

The results of many laboratory and field tests with small engines designed for engine-driven chain saws (Ref. 22) showed that the variation in deposits caused by different oils and different additives is enormous and that a very large number of tests showed almost complete inconsistency. It appeared reasonably certain that for these engines in chain-saw service a mildly detergent oil was preferable to heavily compounded oils and straight mineral oils. Various oils can be roughly evaluated by observing the piston skirt on the exhaust side through the exhaust ports, without disassembling the engine. Oils which give a clean piston skirt on the exhaust side also give minimum deposits on the piston crown, cylinder head, and exhaust passages, and show less tendency to cause ring sticking. Repeated tests have shown that when an oil is used which starts to dirty the piston skirt, a change to a clean-running oil will gradually clean the skirt, although most exhaust deposits will remain. Lower viscosity oils leave fewer combustion chamber and exhaust port deposits, but oils of SAE 30 to 70 seem to be required to provide adequate lubrication in two-cycle engines. Thinner mixtures of oil and gasoline also leave fewer deposits, but again one part of oil for about 16 to 20 parts of gasoline seems to be about the least oil that can be used in most engines. Other engine tests have revealed that if the piston speeds and bearing loads are kept quite low, gasoline-oil mixes of 100 to 1 may be made to lubricate satisfactorily for periods up to 500 hours of continuous operation.

Dynamometer tests of larger engines with a wide variety of oils and under diverse operating conditions (Ref. 23) indicate that viscosity is the only significant property of oils which relates directly to the engine power, friction, and fuel economy. Popular types of "oiliness" or "friction reducing" additives display no measurable improvement on engine friction, power, or fuel economy over that of the oil to which they are added. Oils with additives to improve their viscosity index have engine viscosities appreciably lower than laboratory viscosities probably because such oils are non-Newtonian and develop considerably less viscosity at high shear rates. As a result, high viscosity index oils may show a significant improvement in engine power and fuel economy.

Less extensive studies of lubricants for miniature engines indicate that the conclusions discussed above regarding oils for larger engines hold true for miniature engines. Engine type and operating conditions are more significant than the lubricant used in establishing the performance of an engine. There does not appear to be any best oil for all applications. Castor oil may offer a certain advantage over mineral base lubricants when used in crankcase-compression two-cycle engines. Some investigators feel that castor oil added to the fuel tends to maintain better lubrication films at higher pressures in actual engine operating conditions, and thus maintains adequate lubrication in many critical instances where mineral lubricants would fail. During project test work it was observed that fewer two-cycle engine connecting-rod bearing failures occurred while operating with castor oil than with mineral lubricants. However, the castor oil does have a tendency to form gummy deposits in the cylinder and will not mix with petroleum fuels without an emulsifier. Various synthetic lubricants appeared adequate but showed no marked improvement over mildly detergent SAE 70 petroleum-base oil for two-cycle engines.

In conclusion, the proper choice of oil for a given engine-generator set and type of service will have to be made after the engine is in a rather advanced state of development and can be tested under conditions similar to those expected in service. A mildly detergent petroleum-base oil with a high viscosity index appears to have desirable qualities for miniature engine-generator sets for many types of service. For two-cycle crankcase-scavenged engines SAE 70 is a good viscosity of oil, while for four-cycle engines using splash lubrication in the crankcase, SAE 20 oil appears to be adequate.

APPENDIX I

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